VALVES, VALVE-GEARS AND VALVE DIAGRAMS

Brown

BY

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PREFACE

About eight years ago the author prepared a set of Notes on this subject and they have since been regularly issued and revised every one or two years in neostyle form. This method of issuing notes is admirable for the purpose of making revisions that appear to be desirable after a course in the class room, and the author would be reluctant to abandon this advantage were it not that the well-established points of the subject in general appear to be in such shape that very little revision has seemed necessary the past few years.

On account of the fact that about twenty per cent. of new material, both in text and illustrations, has been added this summer in the preparation for this book, the author feels that there may be some revision of this new matter desirable after it has been tried out in the class room, and has, therefore, decided to publish the book privately and in small editions until, at least, this new part of the subject shall become as settled as the older part. A further prompting for issuing these notes in book form is the fact that during the past few years there has been a small scattered call from graduates who have not kept or have lost their loose-sheet notes, and also a call from outsiders. Books are more satisfactory in meeting such cases.

Notes on this subject at Stevens Institute were started by Professor Jacobus, and continued by Professors Anderson and Pryor, until the subject came into the writer's hands in 1903. The work thus started was part of a more general course in engine work and consisted principally of notes leading up to the drafting-room course, covering eight problems which are now given at pages 17, 28, 50, 54, 64, 82, 98 and 116. Of these problems, four, comprising the double-ported, Meyer, Corliss and floating valves, have been either largely revised or entirely changed.

The material in this book, aside from the drafting-room problems, has been arranged for class-room and recitation work after extended visits to drafting rooms in which the work in the design of valves and valve gears was being carried on in a practical way, and it is believed that the methods here presented will be found to agree fairly well with general practice.

While an arrangement of material that would best fit in with the general course of mechanical engineering at Stevens Institute has been the principal aim of the author in presenting this work, and while many suggestions from numerous sources, including the works of Zeuner, Bilgram, Auchincloss, Welch, Halsey, Peabody, Spangler and Begtrup, have been adopted, there have been introduced some features that have been original in their conception so far as is known to the writer. Principal among these may be mentioned the introduction of numbers marking the order of the drawing of lines in the construction of valve diagrams in the early exercises, thus requiring synthetic as well as analytic study at the outset of the course; the formula for determining exactly the steam-lap by the Zeuner diagram when port-opening, lead and cut-off are given;

the introduction of preliminary free-hand problems before taking up the regular drafting-room problems; the combining of the valve ellipse with the steam engine indicator-card to determine the steam and exhaust laps, steam and exhaust port openings and lead while the engine is in service, or without removing the steam-chest cover; the method of determining the proper width of cut-off blocks for the Meyer valve; the Corliss valve-gear design, and the condensed arrangement of Auchincloss's method of design of the Stephenson link motion.

Among the recently developed methods of steam control that have been included are the Baker valve-gear for locomotives, the Lentz gear for stationary engines and the Curtis and Westinghouse gears for steam turbines.

FRANKLIN DER. FURMAN.

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VALVES, VALVE-GEARS AND VALVE DIAGRAMS

SECTION 1.—SIMPLE STEAM-ENGINE.

The subject of valves and valve-gears embraces all the mechanism of a steam-engine which is employed in automatically regulating the admission and exhaust of steam to and from an engine cylinder.

Elements of Valves and Valve-Gears.

Names of Engine Parts.

The elementary parts of a steam-engine are diagrammatically shown in Fig. 1, as follows:

A, A' is the engine cylinder, B the piston, C the valve, D the piston-rod, E the connecting-rod, F the crank, G the main- or crank-shaft, H the eccentric-sheave, J the eccentric-strap, L the eccentric-rod, and K the valve stem.

Point e is the pin of the cross-head which travels back and forth between two straight guides not shown; d is the crank-pin; a is the center of a circular disc called the "eccentric-sheave," which is keyed to the shaft; $b\,a$ is the eccentric radius and is equal to ½ the travel of the valve, the dotted circle being the path of the point a, g the "bridge-wall," and h the "valve-seat."

Crank End, Head End, Forward Stroke, Return Stroke, Dead-Center.

Before explaining the operation of the engine some of the terms and expressions will be pointed out:

The "crank end" of a cylinder is the end nearest the crank shaft. The "head end" is the end farthest from the crank shaft. The "forward stroke" of an engine occurs while the piston is moving toward the crank shaft; the "return stroke" while moving away from it. The engine is said to be on "dead-center" when the crank, connecting-rod and piston-rod are all in the one straight line, as shown in Fig. 1. There are two dead-center positions in each cycle, one being shown in Fig. 1 and the other occurring when the crank has turned 180° from the position shown. No amount of steam pressure on the piston will turn the engine when it is on either dead-center.

"Running Over," "Running Under."

When referring to the direction of rotation of an engine it is customary to speak of it as "running over" or "running under," instead of running clockwise or counterclockwise. The latter terms are often confusing especially in an engine which will be running clockwise to a person standing on one side and counterclockwise to a person standing on the other side.

An engine is said to be "running over" when the crank rises at the beginning of the forward stroke, or, when the top of the flywheel turns away from the cylinder. It is "running under" when the crank falls at the beginning of the forward stroke, or, when the top of the flywheel turns toward the cylinder. Stationary engines are usually designed to run over, while locomotives must necessarily run under, the cylinders being forward. With engines running over, the pressure between the crosshead and crosshead guide, due to the angularity of the connecting rod, comes on the lower side of the crosshead only and on the body of the engine frame directly; whereas in engines running under, the side pressure due to transmission must come on a specially designed guide-part of the engine frame with the pressure upward away from the main body of the frame.

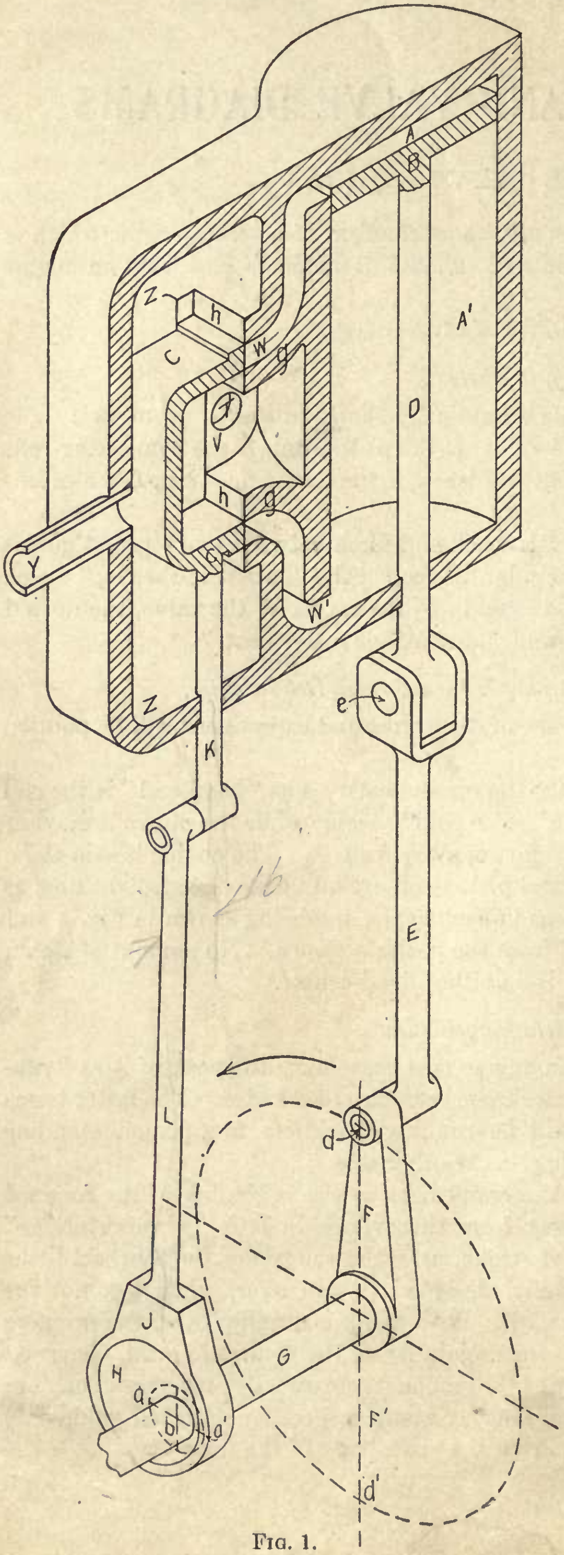


Fig. 1.

Operation of Steam-Engine.

In the working of a steam-engine the parts operate as follows: Steam enters the steam-chest, Z through the pipe Y, Fig. 1. The valve C is moved (downward, for example), and the steam passes through the steam-port W to the cylinder A, thus driving the piston B to the opposite end of the cylinder, and the crank F and the eccentric center a, each through 180° to the positions shown by the dotted lines F' and b a'. During this period of motion in the direction of the arrow, the valve has been at the extreme downward position; it is again central, and is moving upward, and just admitting steam through the steam-port W' to the under side of the piston which is now at the bottom of the cylinder A'. At the same instant the steam-port W is opened to the exhaust-port V, and the exhaust steam on the upper side of the piston escapes through the exhaust pipe T.

Observe carefully that in order to run the engine with *this* valve the effective eccentric-arm a b must be set at 90° with the crank F. The student cannot hope to master this subject without understanding this point thoroughly, and always keeping it in mind.

Elementary Steam Valve.

The valve C, Fig. 1, is of the most elementary form (*i.e.*, the width of valve at seat just equals the width of port), and a study of the figure will show that it admits steam during the entire stroke. Such a valve would be extremely wasteful, for it makes no use of the expansive power of steam. In nearly all engines this elementary valve is modified so as to cut off the admission of steam after the piston has been forced through only a part of the stroke. The piston is then driven through the remainder of the stroke by the expansive power of the steam.

Steam-Lap and Lap Angle.

The modification of the elementary valve necessary to give cut-off at a fraction of the stroke, consists of an addition known as the "steam-lap." In Fig. 2 let the dotted line l limit the edge of the elementary valve; then l m is the

steam-lap. As in Fig. 1 the engine is on dead-center, and the slightest movement of the valve downward will admit steam and drive the piston, assuming of course that the engine has sufficient momentum to pass dead-center; but the valve itself, in Fig. 2, is not central (with respect to the steam-ports) for the dead-center position of the engine. When the lap *l m* was added, the eccentric-sheave was unkeyed and the effective eccentric-arm moved from *b a* to *b k* (while the crank *F* remained stationary), so as to make the distance *c d* equal to the lap *l m*. The angle *a b k* is called the "lap angle."

Lead, Lead Angle, Angle of Advance.

In Fig. 2 the valve is set so as to admit exactly at the end of the stroke. In practice, steam is usually admitted to the cylinder just before the end of the stroke. If now the eccentric is turned still further (from *b k* to *b e*) while the engine remains on dead-center, the edge *m* of the valve will be drawn a small distance (equal to *f c*) across the port *W*. This distance is called "lead," and in small engines is about 1/16 inch. The angle through which the eccentric is thus turned (angle *k b e*) is the "lead angle." The lap angle plus the lead angle equals the "angle of advance" (*a b e*). The total angle by which the eccentric precedes the crank in simple cases equals 90° plus the angle of advance. This entire angle is termed by some as the "angle of advance," to the confusion of the subject unfortunately. The majority, however, define angle of advance as given above, and as so defined is more convenient in the use of valve diagrams and the study of the subject generally.

When the eccentric center is at *k*, Fig. 2, and turning in the direction of the arrow, the edge *m* of the valve is moving downward, and admission of steam to the cylinder begins, assuming zero lead. When the eccentric center is at k' ($< n\ b\ k' = < a\ b\ k$) the edge of the valve is again over the edge of the port *W*, but is now moving upward, and admission ceases. Admission, therefore, has taken place while the eccentric and main shaft have turned through the angle,

$$k\,b\,k' = 180° - 2\,a\,b\,k \quad . \quad . \quad . \quad . \quad . \quad (1)$$

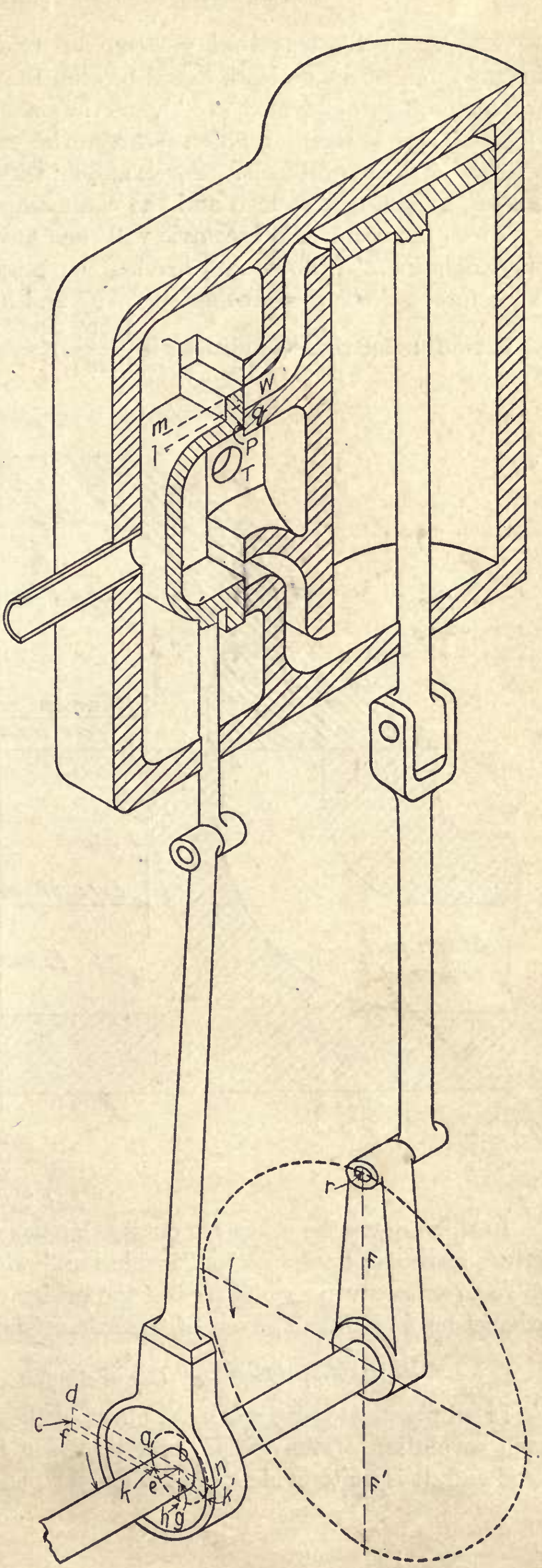


Fig. 2.—Showing valve with steam-lap and eccentric set with lap angle.

The half valve travel $h\,b$ = steam-lap ($c\,d = b\,g$) + steam-port opening ($h\,g$). Considering for the moment a zero lead, it will be seen that the greater the lap, ($l\,m = c\,d = b\,g$), the greater will be the angle of advance ($a\,b\,k$), and the smaller will be the angle $k\,b\,k'$ and the steam-port opening ($h\,g$). There is then a relation between the steam-port opening and the lap which is useful in the solution of later problems. For example: what would be the amount of lap necessary to give ½ cut-off, assuming zero lead and the connecting-rod to be infinite in length?

First, the crank and eccentric will each have turned through 90° when cut-off takes place, and the angle $k\,b\,k'$ will be 90°, leaving the angle $a\,b\,k$ = 45° according to formula (1) on page 1. Therefore, $c\,d = b\,g$ = sine 45° = 0.707 and $h\,g$ = 0.293; or, the ratio of lap to port opening for ½ cut-off under these conditions is $\frac{0.707}{0.293}$

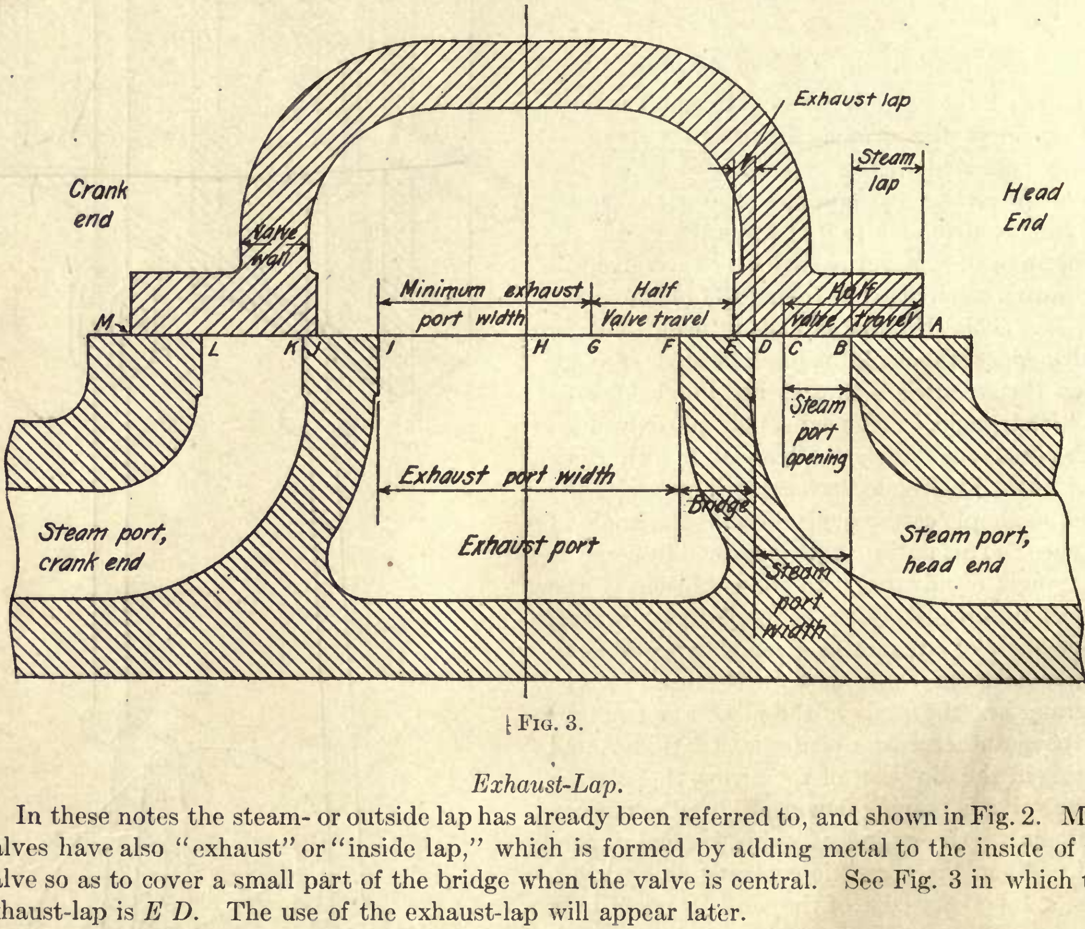


FIG. 3.

Exhaust-Lap.

In these notes the steam- or outside lap has already been referred to, and shown in Fig. 2. Most valves have also "exhaust" or "inside lap," which is formed by adding metal to the inside of the valve so as to cover a small part of the bridge when the valve is central. See Fig. 3 in which the exhaust-lap is $E\,D$. The use of the exhaust-lap will appear later.

Finite and "Infinite" Connecting-Rods—Effect of Angularity in the Finite Rod.

In all practical valve work the effect of the changing angles of the finite connecting-rod during each revolution of the crank must be taken into account. Starting from dead-center position, head end, it is quite evident that when the piston is half-way through its stroke the crank cannot

be exactly 90° advanced; on the forward stroke it will be less than 90°, see angle *a* Fig. 4 and on the return stroke more than 90°, see angle *r*. It will be exactly 90° with the "infinite" connecting-rod, for which there is a mechanical equivalent, (see Fig. 5), which, however, is seldom used. The motion of the point *f* in the "infinite" connecting-rod, Fig. 5, is harmonic, or exactly equivalent to that of the point *c* which is the projection of *b;* whereas in Fig. 6, the point *f* moves faster than *c* while *b* is moving from *d* to *l*, and slower while *b* is moving from *l* to *m*.

The length of the connecting-rod varies in practice from 4 to 8 times the length of the crank for steam-engine work. In this course it will always be taken as 5 times, unless otherwise specified.

The effect of the angularity of the *eccentric-rod* is generally so very small that it is inappreciable, and is therefore neglected. This becomes evident when it is considered that the length of the eccentric-rod is 20 to 30 times the eccentric radius.

In the work of valve design it is necessary to adopt some graphical method which will show at

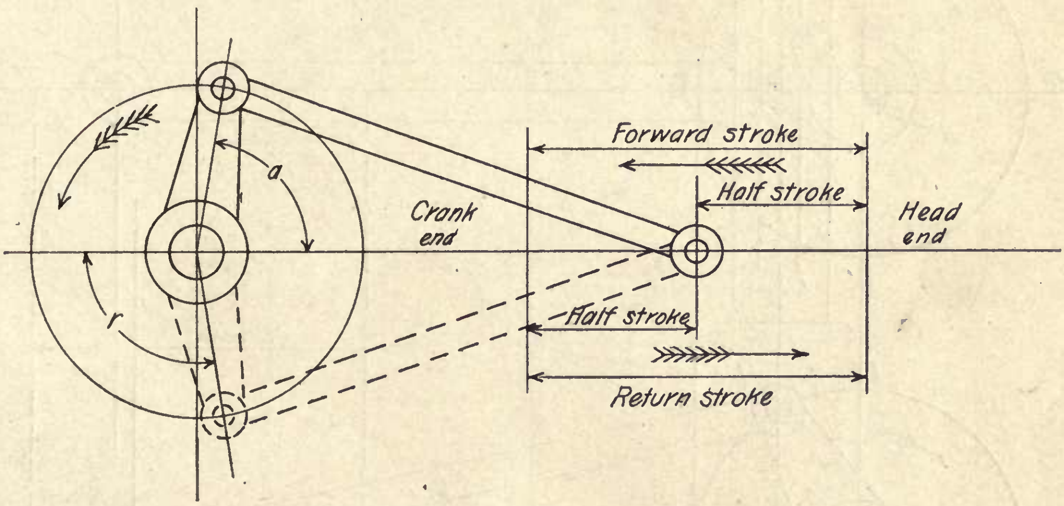


FIG. 4.

a glance the steam distribution at any instant, and also the several positions of the crank at the points of admission, cut-off, release and compression.

In Fig. 2 the valve is shown in position for *admission,* considering that it is moving downward. After traveling to its lowest point and returning to the position illustrated, *cut-off* of the steam takes place and *expansion* occurs. As the valve continues to move upward, *P* reaches *q*, when *release* takes place, the steam exhausting into the exhaust port *T*. The valve then continues to its highest position, and when *P* reaches *q* on the return the steam in the cylinder is trapped for a very short time, during which the piston continues to move upward and *compression* takes place.

ZEUNER DIAGRAM.

Several methods have been devised to show graphically the relative positions of the valve and crank, and the steam distribution, and while each of the more common methods will be briefly described later on, the one to be used in this course will now be taken up. It is known as the "Zeuner diagram." This diagram shows how far the valve is from its central position for any position of the crank. Knowing then the dimensions of the valve and the valve-seat, the actual opening of the port for any crank position, either for entering or exhaust steam, is seen at a glance.

In Fig. 7 let $A\ B$ represent any position of the crank. Then if an angle of advance of 30° be assigned, the eccentric will be 120° in advance of the crank, or in the position $A\ C$.

When it is in the position $A\ C$, Fig. 7, the valve must be off center a distance $C\ L = A\ D$. But $A\ D = A\ E$ since the diameter of the circle $A\ E\ F$ equals the radius of the eccentric circle, and the right-angle triangles $A\ E\ F$ and $A\ D\ C$ are equal.

The radial distance $A\ E$ then represents the amount the valve is off center, and if there were

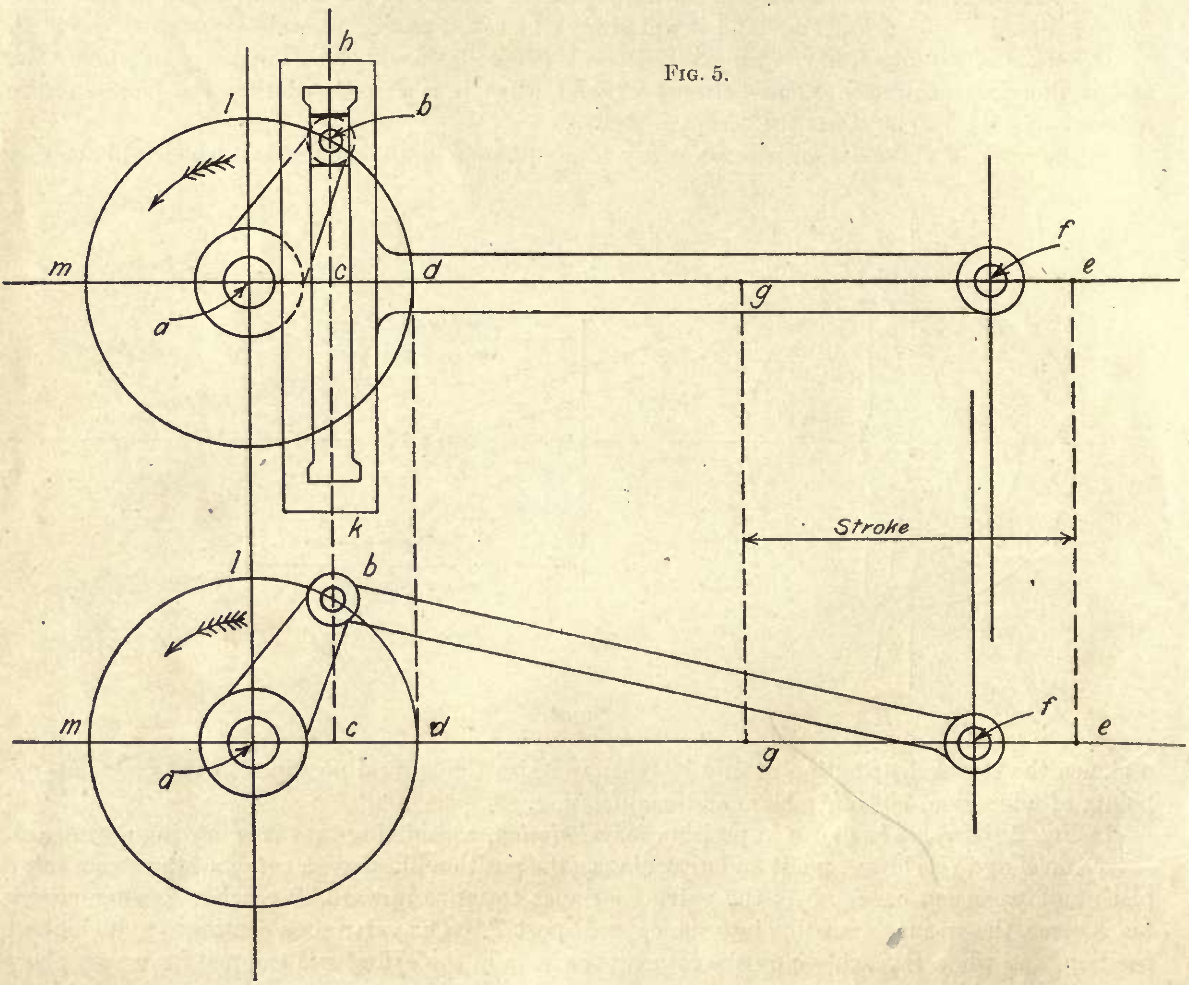


Fig. 5.

Fig. 6.

no lap, as in Fig. 1, it would be the amount of port opening with the crank at $A\ B$. (Fig. 7 it will, be understood, is on a much larger scale than Fig. 1).

Remembering that the eccentric is 120° in advance of the crank, $A\ C$, together with the circle $A\ E\ F$, may be turned back this amount, when $A\ C$ will coincide with $A\ B$, and E will fall at G and the circle $A\ E\ F$ at $A\ G\ H$. The angle $F\ A\ H$, then, is 120°, and taking from it the right-angle $F\ A\ K$ there is left $K\ A\ H = 30°$, which is the angle of advance. *$A\ G$, on the crank-line position,* now measures the amount the valve is off center for that crank position. The circle $A\ G\ H$ when

laid off with the proper angle of advance, may be called specifically the "Zeuner circle," for, no matter where the crank position is drawn, the part lying within this circle always measures the amount the valve is off center.

That $A\,G$ is equal to $C\,L$, Fig 7, and therefore shows the amount the valve is off center may also be shown briefly as follows:

$\angle NAK = \angle MAB$, lines respectively perpendicular

$\angle CAN = \angle KAH$, by construction

$\therefore 90° - (\angle NAK + \angle CAN) = 90° - (\angle MAB + \angle KAH)$ and $\angle CAF = \angle HAB$

$\therefore$ $\triangle$s $A\,C\,D$ and $A\,H\,G$ are right triangles having equal angles at their vertices and equal sides $A\,C$ and $A\,H$. They are therefore equal and $A\,G = A\,D = C\,L$. *Q. E. D.*

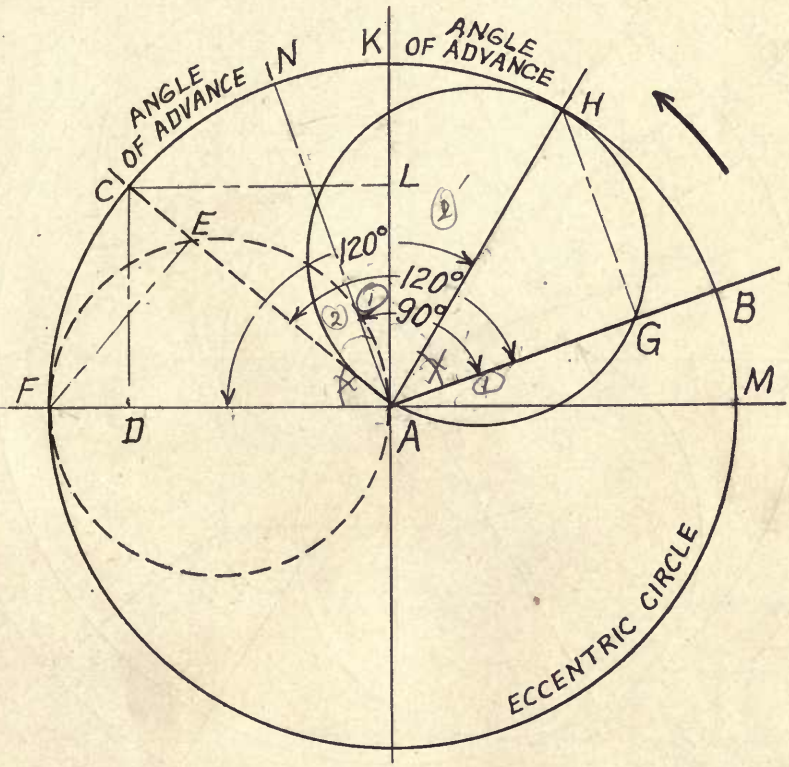


FIG. 7.—Showing Development of Zeuner Diagram.

In using the Zeuner diagram it must be kept constantly in mind that the angle of advance is laid off in the opposite direction from that in which the engine is turning when the eccentric is directly connected to the valve-stem.

Application of the Zeuner Diagram.

Fig. 8 is a practical application of the Zeuner diagram showing the events for the head end steam-port. The throw of the eccentric, the angle of advance, and the steam and exhaust laps are assumed. $A\,K = \frac{1}{2}$ the travel of the valve. $K\,A\,H =$ the angle of advance. When the crank is at $A\,N$ the valve is off center the distance $A\,D$. But $A\,D$ equals the steam-lap, or the distance the valve has to travel from its central position before it begins to open the steam-port.

Therefore the Zeuner diagram shows that steam begins to enter the cylinder when the crank is at *A N*. At the end of the stroke, or on the dead-center position, when the crank is at *A C*, the valve is off center the distance *A B*, and the steam-port is open the amount of the lead = *E B*. With the crank at *A H* the valve is at its extreme left-hand position, and the steam-port is open the maximum amount equal to *F H*. When the crank arrives at *A V P* the steam-port opening is zero, and cut-off takes place. Steam has been admitted then while the crank has been turning from *A N* to *A P*.

When the crank reaches *A T* (tangent to the Zeuner circles) the valve is central, and if there

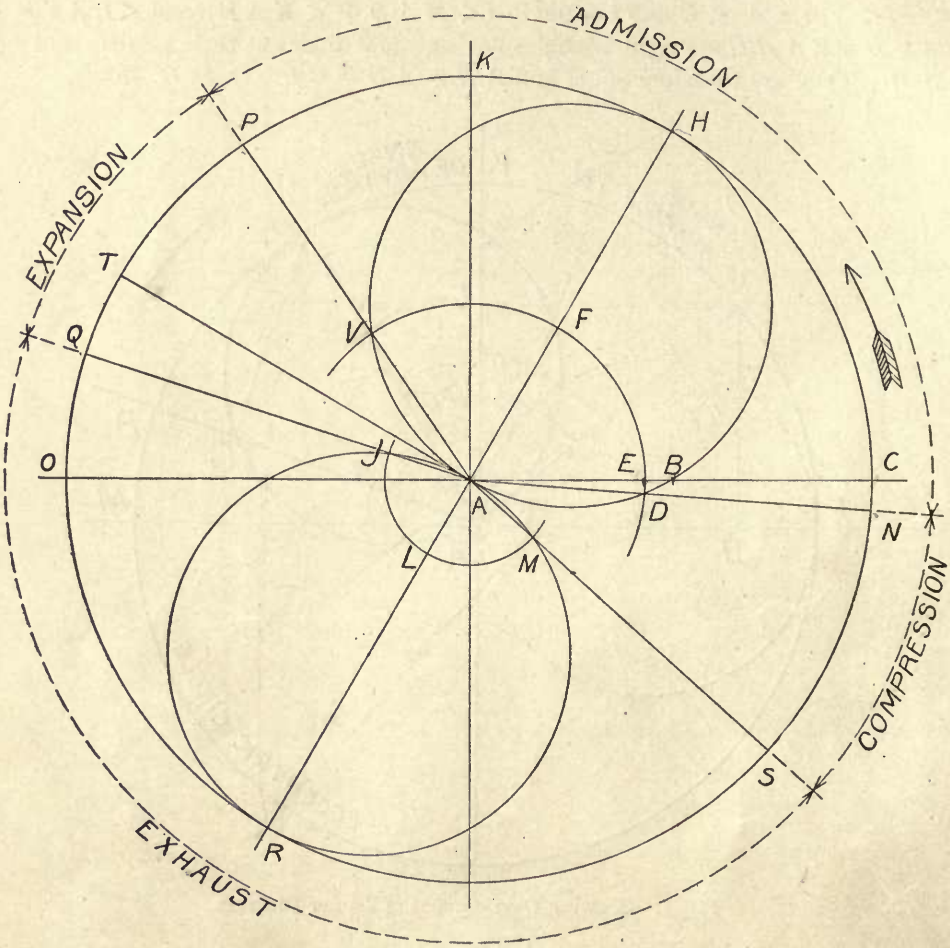


FIG. 8.—Application of the Zeuner Diagram to the Events of the Head End Steam Port.

were no exhaust-lap, exhaust would begin. But in Fig. 8 an exhaust-lap equal to *A L* has been assumed; the valve must therefore move the distance *A L* or *A J*, and the crank reach the position *A J Q* before exhaust begins. The exhaust opening continues to increase until it reaches its maximum, *L R*, at *A R*, and then decreases until it closes altogether at *A S*. The unexhausted steam at that instant is then trapped in the cylinder, and as the piston nears the end of the return stroke, the steam must be compressed until the crank reaches *A N*, when admission again takes place, and the cycle is completed.

The Zeuner diagram showing the events for the crank end port may have the steam-lap arc $D\ F\ V$ continued to intersect the circle $A\ J\ R\ M$, and the exhaust-lap arc $J\ L\ M$ continued to intersect the circle $A\ D\ H\ V$; and crank end admission would begin just before the crank reached $A\ O$.

Principal Phases of a Steam-Engine Cycle.

The four principal phases of the stroke are called "Admission ($A\ N$), "Cut-off" ($A\ P$), "Release" ($A\ Q$), and "Exhaust Closure" ($A\ S$).

Observe, and commit to memory the fact that $\left\{\begin{matrix}\text{steam or}\\ \text{outside}\end{matrix}\right\}$ lap controls admission and cut-off, and that exhaust-lap controls release and exhaust closure.

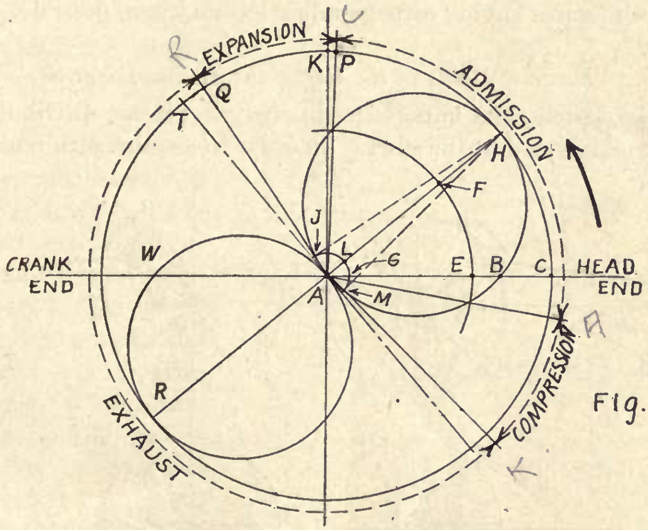


Fig. 9

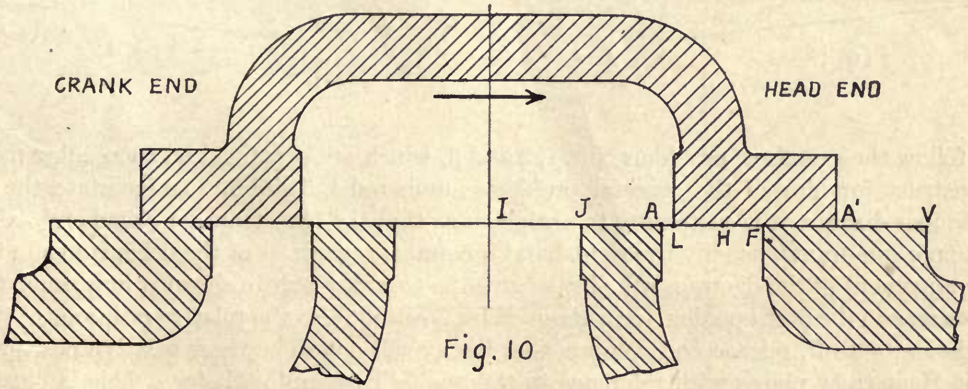


Fig. 10

Positive and Negative Exhaust-Laps.

The exhaust-lap shown by $A\ J$ in Fig. 8 is termed positive exhaust-lap because it represents metal added to the elementary valve, and because the valve has to travel an additional amount beyond its central position to open the port to exhaust steam. But it frequently happens, in order to obtain a more desirable steam distribution to fit special conditions, that the exhaust-lap is decreased in which case it may be zero when the arc $J\ M$ would reduce to the point A, the valve would open

to exhaust just as it reached its central position, and $A\ R$ would be the maximum exhaust-port opening; or, the exhaust lap may be negative, in which case it represents metal cut away from the inside edge of the elementary valve, and the valve opens the port to exhaust before it reaches its central position as shown in Fig. 9.

The negative exhaust lap in Fig. 9 is $A\ L$; exhaust begins when the crank is at $A\ Q$ and the valve is on center when the crank is at $A\ T$. The maximum port opening to exhaust would be $R A + A L$ providing the steam-port were that wide. When the distance $R\ L$ becomes greater than the steam-port width the engine is said to have "full exhaust opening." The intercept on the dead-center crank position $A\ O$ between the exhaust lap and Zeuner circles, $W\ G$ in Fig. 9, is sometimes called the "exhaust lead." A valve having negative exhaust lap equal to $A\ L$, Fig. 9, is illustrated in Fig. 10, to ½ size, both figures having corresponding letters where possible. The negative exhaust lap is shown at $A\ L$.

Exercise Drills in the Use of the Zeuner Diagram.

A valve diagram is of such great importance in analyzing steam distribution for a valve, that it should be thoroughly understood at the start. In order to acquire such understanding the student

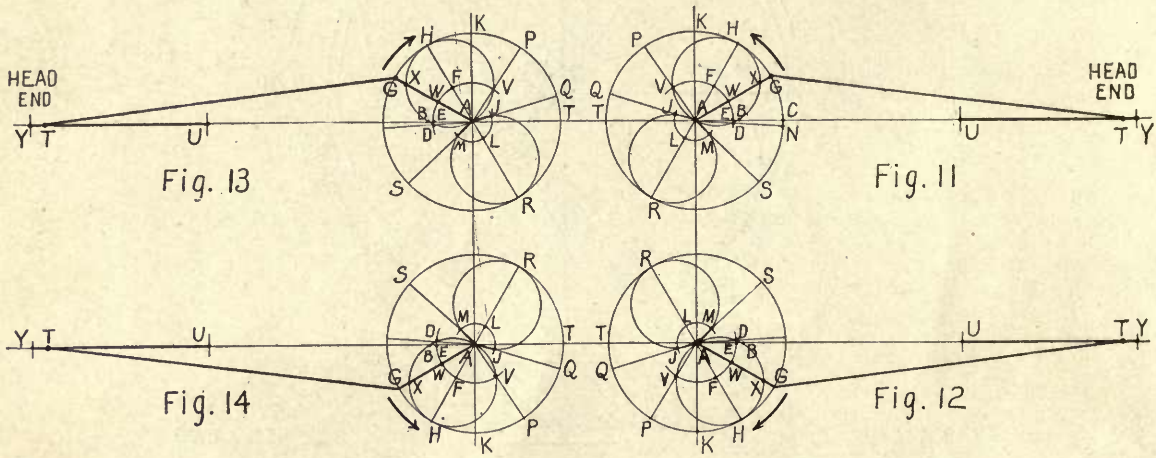


Fig. 13 Fig. 11

Fig. 14 Fig. 12

should follow the exercises, problems Nos. 1, 2 and 3, which are explained on succeeding pages, and also construct for himself the original problems numbered 4, 5 and 6. To facilitate the drawing of the Zeuner diagram it is customary to arrange it so that the live steam-lap, head end, falls in one of the upper quadrants, usually the right-hand quadrant, regardless of the other items in the data. This arrangement of the diagram will always give the correct result in showing how much the valve is off center, and the port opening, for a given crank position; also the relative positions of the crank at admission, cut-off, release and compression, but it will not show these relative positions of the crank in their right places with reference to the engine base and cylinder. This is illustrated in Figs. 11, 12, 13 and 14, where it will be seen that for the crosshead position T, which is about .08 on the forward stroke, in each case, that the valve is off center a distance $A\ X$ and the port open an amount $W\ X$, and these values are equal in all four figures. Also the crank positions for all events are the same distances from each other in all four figures. When the data specify only crank positions it is not necessary to draw the crosshead position and the diagram may be arranged as in Fig. 11, but when the piston positions are specified it is more satisfactory to arrange the diagram so the resultant crank positions will show in their right places relatively to the engine frame.

1. Given: Eccentricity, crank position at cut-off, angle of advance, and compression.

Find: Steam-lap, exhaust-lap, crank positions at admission and release, lead, exhaust-lead, and greatest steam and exhaust port openings.

The solution of this problem is shown in Fig. 15, the data being in dash-line construction, and the rest of the work in solid-line construction. The order of drawing all the lines is shown by the numerals on each. The steam-lap is *o g;* exhaust-lap, *o t;* crank position at admission, *o r*, and at release *o q;* lead, *h k;* exhaust lead, *m n;* greatest steam opening, *s c;* greatest exhaust port opening *t d*.

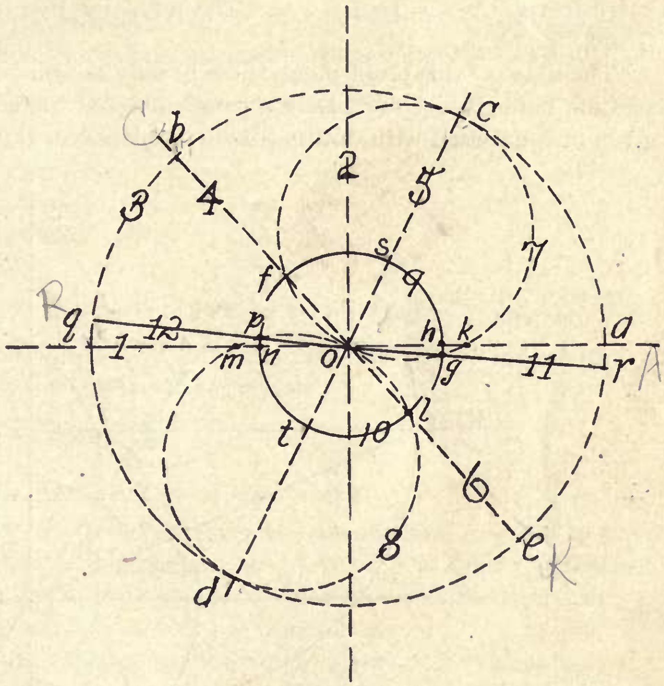


FIG. 15.

2. Given: Steam-lap, lead, crank position at cut-off, and exhaust-lead.

Find: Valve-travel, angle of advance, exhaust-lap, and crank positions at admission, release and compression.

The data are given in Fig. 16 in dash lines and by the distances *b c* and *g h*. The results are: *e d* = valve-travel; *m o d* = angle of advance; *o k* = exhaust-lap; *o f*, *o k* and *o l* = crank positions at admission, release and compression respectively.

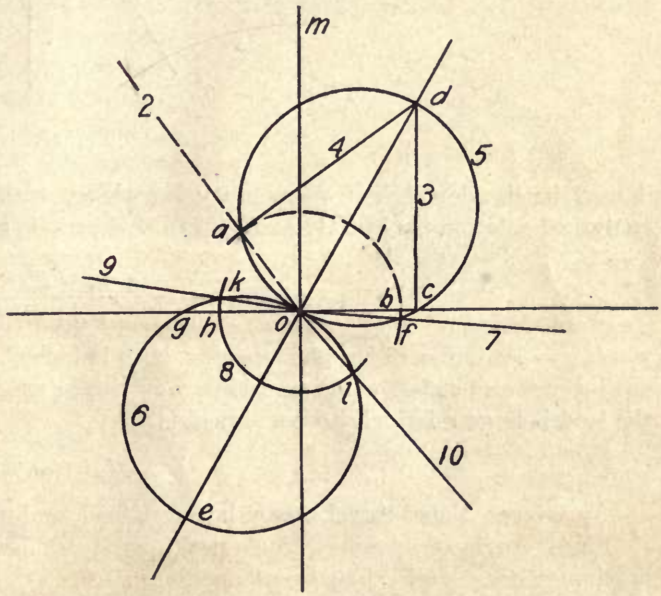


FIG. 16.

3. Given: Cut-off, lead, and steam-port opening.

Find: Lap, valve-travel, and angle of advance.

In Fig. 17, draw given crank cut-off position *o t*, and on *o t* extended lay off *o a* = lead to enlarged scale. Make *a b* = given steam opening to same enlarged scale. Then draw *b u* parallel to line of stroke, and make *b c* = *b a*. Draw *o c* and on it lay off *o d* = *o a*, and draw horizontal line *d f*. With radius *o e* (where *u b* crosses vertical center line) draw circular arc *e v*, and lay off arc *f h* = arc *g e*. Draw line *h w*, which will contain the diameter of the Zeuner circle and *y o w* will be the angle of advance. Take any point as *j* as center for a trial Zeuner circle. This gives a port opening of *z k*, and lead of

$l\,m$, which are too small each in the same proportion. Since $z\,k$ is the trial port-opening and $a\,b$ the desired opening, draw $k\,q$ in any direction, lay off $k\,n = a\,b$, draw $z\,n$, and $o\,i$ parallel to $z\,n$. Then $i\,n$ = radius of desired lap circle, and $k\,i = w\,o$ = diameter of the Zeuner circle or ½ valve travel. $p\,x$ = lead = $o\,a$. The proof for this construction is given in Spangler's "Valve Gears," p. 22.

The data for this problem are those usually assigned in practical work. The involved graphical solution here given is not easily remembered and therefore not generally used, the simple method given in connection with drafting-table problem No. 1, p. 17, being the one usually employed since

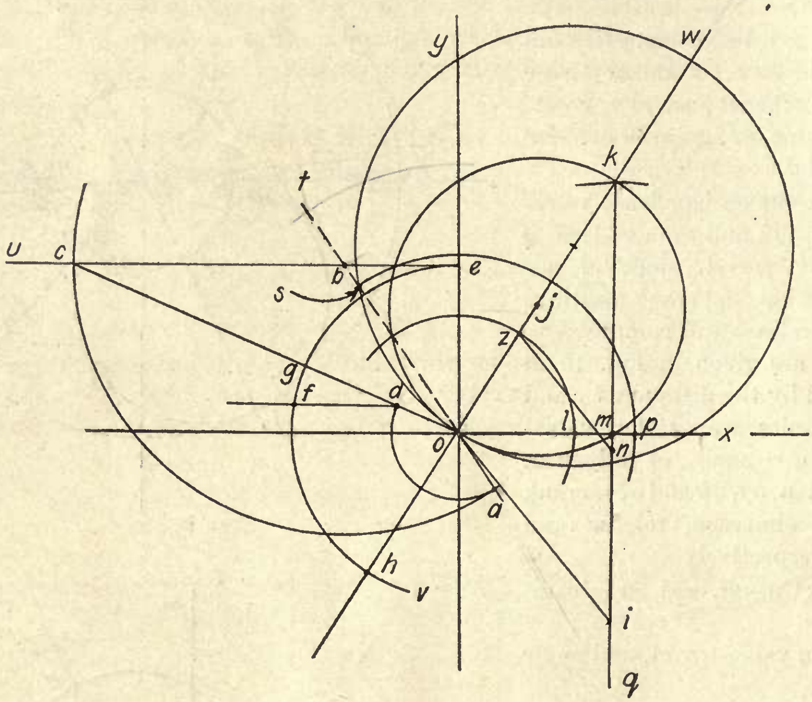


FIG. 17.

it may be developed from elementary knowledge without reference to any book or notes. An analytical solution by Mr. G. A. Pfeiffer, M.E., (Stevens '10) using the formula,

$$\text{Steam-lap} = \frac{2b - a + \sqrt{2b\,(b - a)\,(1 + d)}}{1 - d} - b$$

where a = lead, b = steam-port opening, both in inches, and d = cosine of angle measured between dead-center and cut-off crank positions, may also be used. It is of special advantage in cases where the lead is large relatively to port-opening.

Exercise Problems.

4. Given: Valve-travel, steam-lap, zero lead, and negative exhaust-lap.

Find: Angle of advance, crank positions at admission, cut-off, release and compression, also maximum steam and exhaust-port openings.

5. Given: Crank positions for admission and cut-off on head end, and for admission on crank

end; also, valve-travel, and positive exhaust-lap on head end, and negative exhaust-lap on crank end.

Find: Steam-lap for both ends, crank positions at release and compression for both ends, cut-off crank end, lead for both ends.

6. Given: Angle of advance, valve-travel, negative lead, and crank position at compression.

Find. Steam- and exhaust-laps, and crank positions at admission, cut-off and release.

Steam-Pipes, Steam-Ports and Steam-Port Opening.

The areas of the steam-pipes, ports and port-openings depend on the size of the cylinder and the speed of the piston.

Let L be the length of the piston stroke, in feet,

D the diameter of the cylinder, in inches, and

N the number of revolutions per minute.

Then the volume swept through by the piston per minute will be represented by $2\,N\,L\frac{\pi D^2}{4}$. The velocity V (in feet per min.) of the steam through the port-opening, multiplied by the area A (in sq. ins.) must be equal to the above expression, thus giving the equation

$$V\,A = 2\,N\,L\frac{\pi D^2}{4}$$

Distinction Between Port Width and Port Opening.

Good average practice in a large number of engines allows a velocity of 6,000 to 8,000 feet per minute for the supply steam-pipe, and for the steam-port opening. A velocity of 4,000 to 6,000 feet per minute is allowed for the exhaust through the steam-port. The distinction between "steam-port" and "steam-port opening" should be carefully noted. It is as follows:

Since only one port leads to one end of the cylinder in the simple engine, it is evident that the supply or live steam must go in, and the exhaust steam come out of the same port. The allowance usually made for the velocity of the exhaust steam regulates the area of the steam-port which must be entirely uncovered by a sufficient travel of the valve when opening to the exhaust port. When the valve opens the steam-port for the admission of live steam, it is evident that the entire area of the steam-port need not be uncovered, on account of the live steam flowing at a faster rate than the exhaust. The amount that is uncovered is called the "steam-port opening." This opening is usually less than the width of the port, but in some engines other considerations control the design, and the edge of the steam-lap of the valve may not only cross the entire width of the port, but also traverse a small part of the bridge.

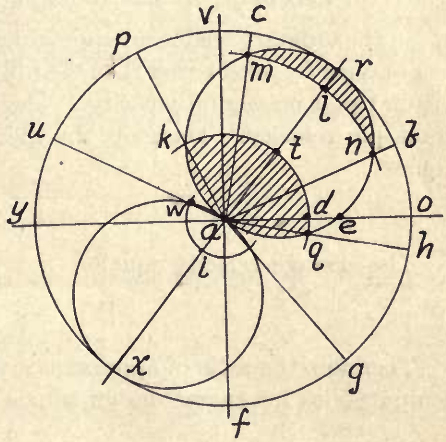


Fig. 18.

Overtravel

The amount that the cut-off edge of the valve travels beyond the live-steam edge of the bridge is termed by some as "overtravel"; whereas it appears more logical to term that travel of the valve beyond the point necessary to give the full calculated steam-port opening as the overtravel, and this word will be used in the latter sense throughout this book. As an illustration, refer to the Zeuner diagram, Fig. 18. There it will be assumed that it has been expedient to make the half-valve travel equal to $a\,r$, whereas the calculated steam-

port opening turns out to be $t\ l$; $l\ r$, then, becomes the overtravel. This overtravel would be represented on the valve seat itself by the amount A travels beyond C in Fig. 3, p. 4. In an engine already built the point C, of course, would not be in evidence, but in order to take intelligent action with respect to the valve or valve-gear, it would be necessary to make computations as to port-opening, etc., thus locating C after which the overtravel could be readily determined. If the overtravel should be great enough to come too close to the exhaust side of the bridge alterations must be made either in the valve-travel or the bridge thickness; this case will be taken up in connection with drafting-table problem No. 1, p. 17.

Formula for Calculating Live and Exhaust Steam-Ports.

In writing the formula $V\,A = 2\,N\,L\frac{\pi\,D^2}{4}$ the symbol a is usually substituted for $\frac{\pi\,D^2}{4}$, and A and a are expressed in square inches, while L and V are given in feet. The equation should then be written $12\,V\,A = 2 \times 12\,L\,a\,N$, or cancelling and transposing, $A = \frac{2\,L\,a\,N}{V}$.

Actual and Average Velocity of Flow of Steam Through Ports.

In building up this formula, it will be observed that the quantity of steam per minute was based on an assumption that steam entered the cylinder during the entire stroke. Inasmuch as engines cut off anywhere from ¼ to ¾ stroke, as a rule, this may seem a needlessly large assumption; but it must be remembered that it is the rate at which steam is required at a given instant that counts, and not the period during which it is required. Owing to the varying velocity of the piston, the rate of flow of 6,000 ft. per minute here provided is only an *average rate,* and means nothing so far as actual rate of flow through the ports at any given instant, is concerned. It is purely an empirical value based on practical experience.

To find the actual velocity of the steam through the ports for any given engine or any given design, it would be necessary to find the piston velocity and the port-opening at successive intervals from which the actual rate of flow through the port at these phases could be determined, the area of the piston being known. If these values were plotted as ordinates, a curve would be obtained in which the maximum ordinate would give the maximum rate of flow of steam through the ports.

In the ordinary engine an approximation to the maximum steam velocity through the ports may be obtained by considering that the full steam-port opening area is uncovered at the instant that the piston has it maximum velocity. The maximum piston velocity V_1 is equal, approximately, to the crank-pin velocity. Therefore the approximate maximum steam velocity through the port opening equals

$$V_1 = \frac{2\,\pi\,R\,N \times \frac{1}{4}\,\pi\,D^2}{A}$$

The average velocity equals

$$V = \frac{2\,L\,N \times \frac{1}{4}\,\pi\,D^2}{A}$$

Therefore the ratio of approximate maximum velocity to the average velocity ordinarily used in computations for engine design equals

$$\frac{V_1}{V} = \frac{2\,\pi\,R\,N \times \frac{1}{4}\pi\,D^2}{A} \times \frac{A}{2\,L\,N \times \frac{1}{4}\,\pi\,D^2} = \frac{\pi R}{L}$$

But since $L = 2R$,

$$\frac{V_1}{V} = \frac{\pi}{2}$$

The average steam velocity, V, in the above formula, allowed by builders of different types and sizes of engines, varies widely, and instead of 6,000 to 8,000 feet per minute, 10,000 and even more is sometimes used.

After the area of the port has been calculated, a length must be assumed in order to determine the breadth. In plain slide-valve engines the length of the port varies in practice from ⅝ to ⅞ of the diameter of the cylinder.

Note Book Problems.

The problems here given should be carefully worked out in a large note book or on large pad paper. They are preliminary to the drafting-table problems which follow.

Prob. 1. Construct on large scale, a valve and valve-seat with assumed values for the live steam and exhaust steam-laps, the bridge, the exhaust-port, the steam-ports, the steam-port opening, and the half valve-travel, all plainly marked.

Prob. 2. Make orthographic drawing on enlarged scale of the lower part of Fig. 2 of this book, assuming the angles *a b k* and *k b e*, the eccentric center at *e*, and engine on dead-center. In determining the angle made by the center-line of eccentric-rod with the center-line of the engine, the eccentric-rod length may be taken equal to 20 × eccentric radius. Mark plainly by use of reference letters:

(1) Lap angle. (3) Angle of advance. (5) Lead.
(2) Lead angle. (4) Steam-lap. (6) Port-opening.
(7) Half valve-travel.
(8) Angle through which crank turns while the piston is moving from end of stroke to point of cut-off.
(9) Angle through which the crank turns while steam is being admitted.

Prob. 3. Construct, to full size scale, an eccentric-sheave, eccentric-strap, and part of eccentric-rod that will give a 2″ valve-travel when mounted on a 2½″ shaft.

Prob. 4. To show the variable motion of the piston during forward and return strokes caused by the angularity of the connecting-rod when the center-line of the stroke passes through the axis of the shaft.

Draw to scale a crank, a connecting-rod, and the cross-head travel, making the connecting-rod equal to 4 crank lengths. Assume the crank length.

From the drawing, fill in the blank spaces in the following items:—

(1) When the piston is at ½ the *forward stroke* the crank has turned through........degrees approximately.

(2) When the piston is at ½ the *return stroke* the crank has turned through........degrees approximately.

(3) When the crank has turned through 90° on the *forward stroke* the piston is at........% of its stroke approximately.

(4) When the crank has turned through 90° on the *return stroke* the piston is at........% of its stroke approximately.

(5) The maximum angle of the connecting-rod is........degrees approximately.

Prob. 5. To show the variable motion of the piston during forward and return strokes caused by the angularity of the connecting-rod when the center-line of stroke is tangent to the crank-pin circle.

Make drawing to scale using same dimensions for crank and connecting-rod as in Prob. 4, and fill in the blank spaces in the following items:

(1) The piston travel = ... × crank length.

(2) The crank travel = ... degrees approximately on the *forward stroke.*

(3) The crank travel = ... degrees approximately on the *return stroke.*

(4) The piston pressure is transmitted without angularity of the connecting-rod and with maximum crank leverage when the piston is at .... % of its forward stroke. This, together with the fact that the angularity of the rod varies from a minimum to a maximum of .... to .... degrees during the return stroke makes it useful only for single acting engines.

Prob. 6. Determine the ratio of lap to port-opening for 0.7 cut-off and zero lead;

(1) For connecting-rod = 5 crank lengths,

(2) For infinite connecting-rod.

In drawing make separate crank and eccentric circles.

Prob. 7. Determine the ratio of lap to port-opening for 0.4 cut-off and zero lead, for a connecting-rod equal to 4 crank lengths.

Prob. 8. Find the maximum rate of flow of live steam through the port-opening of an engine having 10″ bore, 18″ stroke and 250 r.p.m. in which the port-opening has been designed for an average rate of flow of live steam of 6,000 ft. per minute.

Prob. 9. Given: valve-travel = 3″, angle of advance = zero, steam-lap = ½″, exhaust-lap = ¼″, steam-port width = 1½″, bridge ¾″, and exhaust port = 2½″. Find the crank positions for admission, cut-off, release, and compression. Construct the valve-seat, and the valve in its proper position for the beginning of the stroke. Indicate the maximum steam-port opening, exhaust-port opening and lead, both on Zeuner diagram and valve-seat. Then assume any crank position and dot the corresponding position of valve on the valve-seat and mark the port-opening for that position on both the Zeuner diagram and valve-seat.

Prob. 10. Let the data and requirements be the same as the previous problem, only changing the angle of advance to 30°. Take the assumed crank position in the same place as in Problem 9.

Prob. 11. Show effect of changing conditions as indicated in the first column of the following table, on the time when the principal events of the stroke occur, based on a study of Probs. 9 and 10. Fill out the following Table:

	Admission.	Cut-off.	Release.	Exhaust closure.
Increase in Angle of Advance.				
Increase in Valve-Travel.				
Increase in Steam-Lap.				
Increase in Exhaust-Lap.				

DRAFTING TABLE PROBLEM, NO. 1.—PLAIN D-VALVE.

Design a slide-valve for an engine having . .9. bore and. .16. stroke, running at 150. . .revolutions per minute. Cut-off at 75. . .stroke head end; release at. 95. .stroke both ends; lead $\frac{1}{16}$". .head end; average velocity of live steam through ports 100 feet per second; length of connecting-rod 4 times the crank.[1] Average velocity steam thr. exhaust = 66 $\frac{2}{3}$'/sec

equally, compression

Construction of Zeuner Diagram.

Calculate the area of the *port-opening* for the given steam velocity. Make the length of the port 0.7 of the bore, and determine the width of the port-opening.

By means of the Zeuner diagram the necessary steam-lap, exhaust-lap, the valve-travel and the angle of advance may be found as follows:

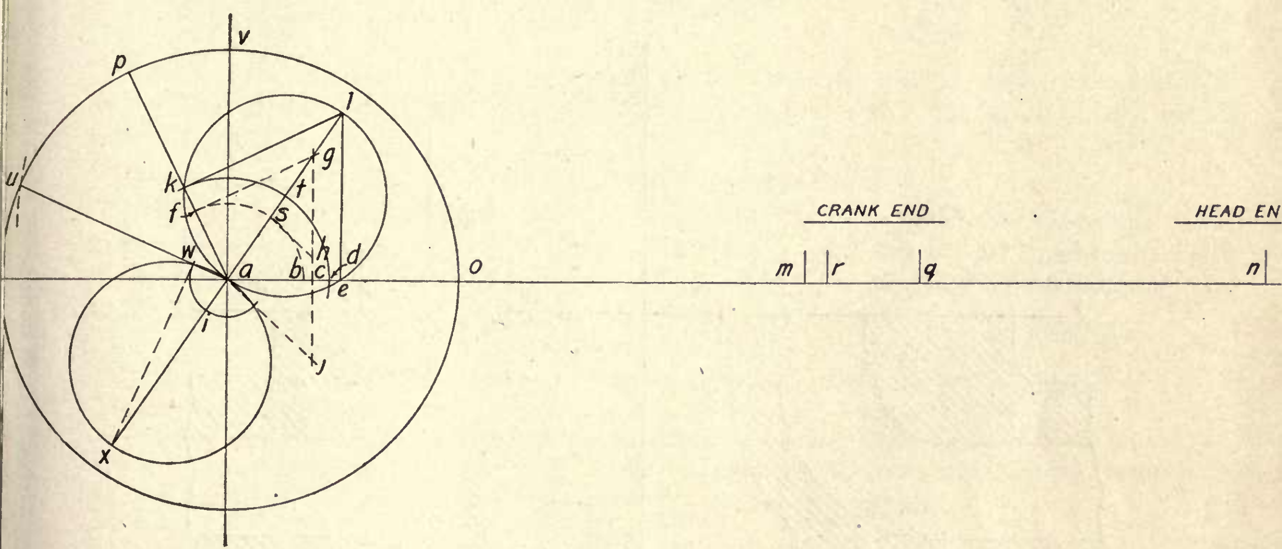


FIG. 19.

On the horizontal line *y z*, Fig. 19, set off *a o* equal to the crank length , *o n* equal the length of the connecting-rod, and *n m* equal to the stroke, to the largest scale that the drawing paper will accommodate. In doing this consider *m n* the cross-head travel instead of the piston travel. *o p y* is then the crank-pin circle, and *a* the center of the crank-shaft.

Find the position *a p* of the crank for the assigned cut-off. A trial steam-lap for the head end of the valve which will give this cut-off approximately should then be found by means of the relation existing between port-opening and steam-lap as explained on pp. 3 and 4 of this book.

Inasmuch as a definite amount of lead is assigned in this problem the ratio of lap to port-opening, just referred to, will not be exact in this case. It will, however, be approximately correct, and will serve as a guide in obtaining the exact lap as follows: At the intersection of the trial lap circle, *a f*, Fig. 19, with the cut-off crank line *a p*, draw a perpendicular *f g*. From *b* where the trial lap circle intersects the engine center-line, lay off distance *b c* equal to the lead on the scale adopted for the trial lap circle. At *c* erect a vertical line until it meets *f g*. Then if the radial distance *s g* equals the cal-

[1] The above data for the cut-off and release refer only to the head end of the cylinder. After the dimensions for the head end of the valve have been found according to the following directions, instruction regarding the crank end will be given.

culated width of the port-opening according to the adopted scale, the assumed lap is the correct one. It is not to be supposed that one will assume the correct lap circle the first time, in which case proceed as follows:

On the line *g c* lay off *g h* equal to the calculated width of the port-opening, and draw *s h*. From *a*, draw *a j* parallel to *s h* until it meets *g c* produced at *j*. Then *h j* will be approximately the length of the required lap, and may be used for the radius of the new lap circle *k t d*. Find point *l* in the same manner that *g* was found. Then *t l* should equal the calculated width of the steam-port opening within $\frac{1}{64}''$. If it does not, a third proportion based on the second must be made.

Then *a t* is the required lap for the given cut-off, *t l* the maximum width of the steam-port opening, *a l* the half-travel of the valve, and *v a l* the angle of advance. To find the exhaust lap that will give release at the assigned time, locate the crank position *a u* for the cross-head at *r* (*n r* = the

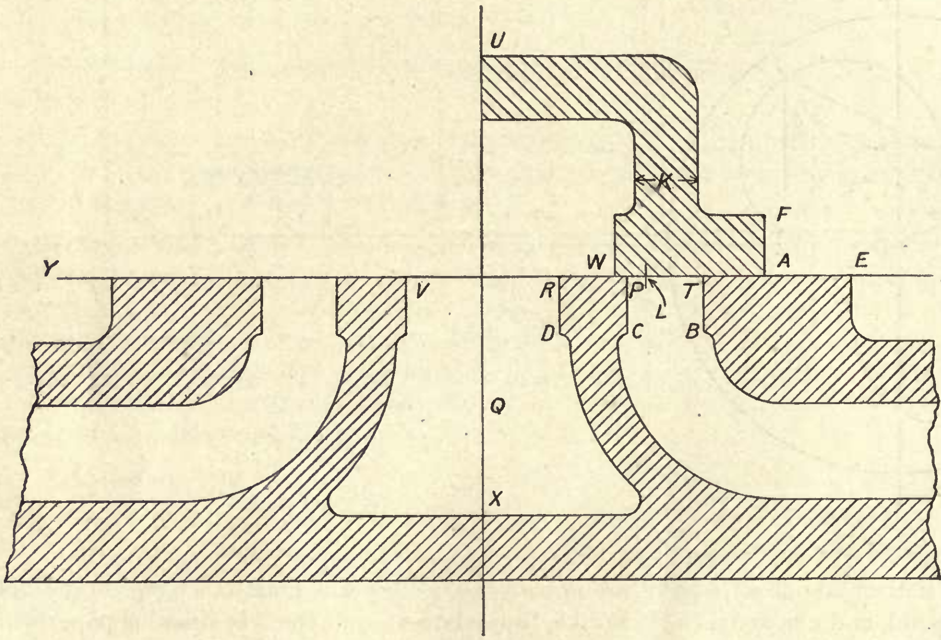


FIG. 20.

given percentage of stroke). *a u* intersects the Zeuner circle *a x* at *w*, and *a w* is therefore the required exhaust lap. The point *w* is determined exactly by drawing from *x* a perpendicular to *a u*. If the crank position *a u* had intersected the Zeuner circle *a l*, the exhaust lap would have been negative; that is, with the valve in its central position the steam-port would be partly open and in communication with the exhaust port.

Layout of Valve and Valve-Seat.

Having all necessary data, the head end of the valve and the ports may now be laid down full size as follows: Draw the valve-seat line *Y Z*, Fig. 20. When completed the valve is to be shown in its central position.

The drawing of the sectional view of the valve and ports (Fig. 20) is to be made full size. (This makes three separate scales to be used in this problem, namely; the crank scale, the Zeuner scale, and the valve scale). Therefore from any convenient point, *A*, on *Y Z*, Fig. 20, lay off *A T* = *a t* of Fig.

19 equal to the steam-lap. Lay off $T\,P$ equal to the calculated width of the steam-port, and $P\,W$ equal to the exhaust-lap as found at $a\,w$ in Fig. 19. $T\,L$ is the maximum steam-port opening, and equals $t\,l$, Fig. 19. $A\,L$, Fig. 20, then equals the eccentricity, or ½ the travel of the valve.

Formula for Minimum Width of Bridge.

The width of the bridge $P\,R$ should in all cases be at least equal to the thickness of the cylinder wall, in order to secure a reliable casting. For an engine of this size this thickness may be ⅝″ to ¾″ according to judgment, and should be taken within this range unless it violates the following standard rule:

$$\left\{\begin{matrix}\text{Minimum}\\ \text{width of}\\ \text{bridge.}\end{matrix}\right\} = \left\{\begin{matrix}\text{Width}\\ \text{of port-}\\ \text{opening.}\end{matrix}\right\} + \text{Overtravel} + \tfrac{1}{4}'' - \left\{\begin{matrix}\text{Width}\\ \text{of steam-}\\ \text{port.}\end{matrix}\right\}$$

This formula will affect the width of bridge only when the edge A of the valve comes within ¼ inch of the edge R of the bridge, and applies principally in repair work. The amount that A travels beyond L is the overtravel.

Formula for Width of Exhaust Port.

$R\,V$ is the width of the exhaust port, and must be so taken that when the exhaust-lap of the valve is in its extreme left-hand position there will still be a width left at least equal to the width of the steam-port. $R\,V$ may thus be determined graphically, or calculated by the following rule:

$$\left\{\begin{matrix}\text{Width of}\\ \text{exhaust}\\ \text{port.}\end{matrix}\right\} = \left\{\begin{matrix}\text{Maximum}\\ \text{inside}\\ \text{lap.}\end{matrix}\right\} + \left\{\begin{matrix}\text{Half-}\\ \text{travel}\\ \text{of valve.}\end{matrix}\right\} + \left\{\begin{matrix}\text{Width}\\ \text{of steam-}\\ \text{port.}\end{matrix}\right\} - \left\{\begin{matrix}\text{Width}\\ \text{of}\\ \text{bridge.}\end{matrix}\right\}$$

In using this formula it must be kept in mind that the exhaust-lap may be different on the two ends of the valve, according to the conditions of the problem, and that therefore the size of both exhaust-laps must be known, and the greater value used in the formula.

Equalizing Cut-Offs by Unequal Steam-Laps.

To determine the exhaust lap on the crank end of the valve it will now be necessary to consider the conditions affecting the crank end, as follows:

If cut-off occurs at the same percentage of the forward and return strokes it is said to be equalized. On the Zeuner diagram, already used, locate the crank position for equalized cut-off on the crank end, and dot in the corresponding lap circle. The diagram will now show that equalized cut-off obtained in this way gives excessive lead on the crank end, and is, as a rule, impracticable. It will not be used in this problem, but the "excessive lead" thus obtained should be marked as such on the diagram for future reference.

Another method of equalizing cut-off without obtaining excessive lead will be described on a later page.

EQUALIZING COMPRESSION BY UNEQUAL EXHAUST-LAPS—SPECIAL AND GENERAL CASES.

The steam-lap on the crank end of the valve is to be made, in this problem, equal to that on the head end, and the crank and piston positions at admission and cut-off determined. The exhaust-lap already determined for the head end fixes the exhaust closure, or beginning of compression, for that end. Now determine the exhaust-lap that will give the same amount of compression on the crank end as on the head end. Then complete the design of the crank end of the valve as follows:

Having determined *R V*, the center-line *U X* of the valve and ports may be drawn. The area *Q* of the cross-section of the exhaust port may be made equal to or a little less than the area of the steam-port. The edges of the ports, as at *T B*, *P C*, *R D*, etc., are faced surfaces, while the remainder of the port is made a trifle larger, and is rough cast. The valve-seat should be limited, as at *E*, so that the edge *A* of the valve will overtravel ¼″. The thickness *A F* of the lap is generally made about the same as the bridge, and the thickness of the valve wall *K* a little less.

Place the necessary working dimensions on the design and mark the finished surfaces. Tabulate the results as follows:

	Travel.	Lead.	Steam-lap.	Exhaust lap.	Steam-port opening.	Part of Stroke completed when			
						Admission begins.	Cut-off takes place.	Release begins.	Exhaust closure occurs.
Head end.									
Crank end.									

Equalization of Release and Exhaust Closure by Unequal Exhaust Laps.

Special Case.

If a valve were constructed with zero exhaust-lap on each end, release on the head end and compression on the crank end would occur simultaneously when the crank is in the position *o b*

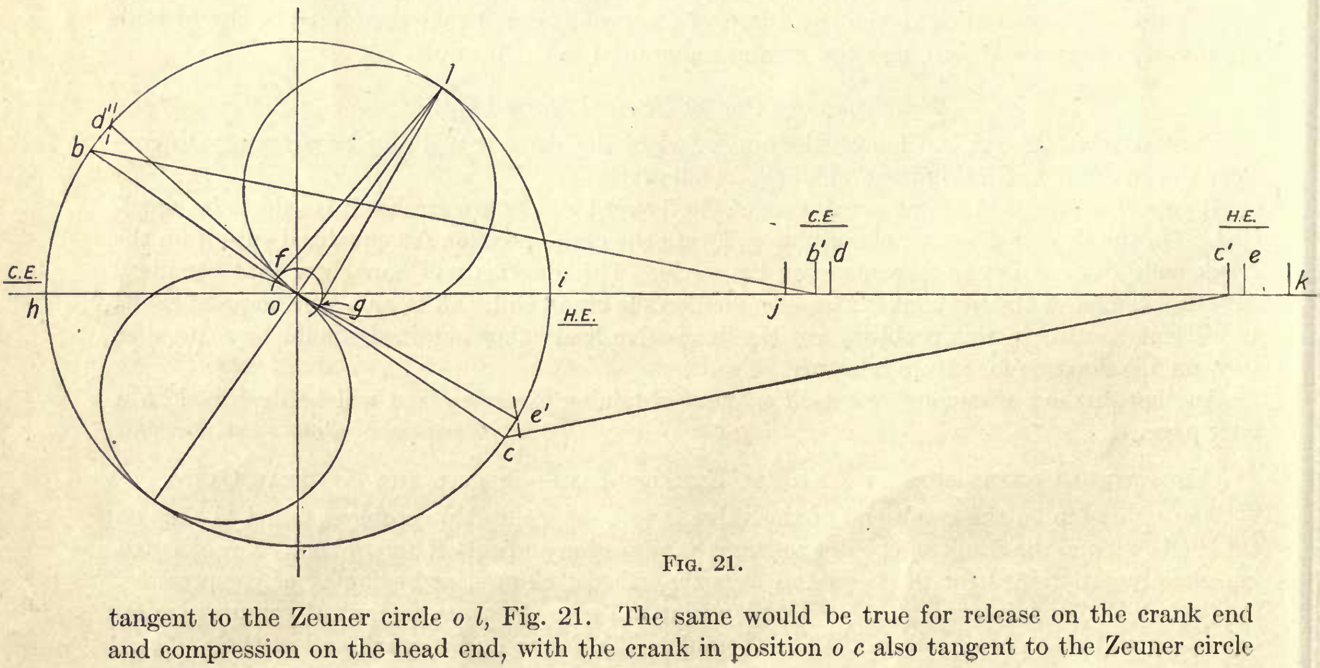


Fig. 21.

tangent to the Zeuner circle *o l*, Fig. 21. The same would be true for release on the crank end and compression on the head end, with the crank in position *o c* also tangent to the Zeuner circle *o l*.

When the crank-pin is at b the piston is at b', and with the crank-pin at c the piston is at c'. $j\ k$ represents the stroke of the piston. $j\ b'$ is smaller than $k\ c'$, and therefore neither release nor compression is equalized on the forward and return strokes when the exhaust-lap on both sides is zero, and indicator cards from the two ends of the cylinder will not be similar.

In order to equalize these events, assume that release on the head end card is desired when the piston is at d, and compression when at e; also that release is desired on the crank end card when the piston is at e, and compression when at d. Then since $j\ d$ equals $k\ e$, release and compression will be equalized on both cards.

With d as a center and a radius equal to the connecting-rod, locate the crank-pin center d' corresponding to d; locate similarly, e'. By drawing the crank position $o\ d'$, we find that the valve requires a negative exhaust-lap equal to $o\ f$ on the head end; similarly, a positive exhaust-lap equal to $o\ g$ is required on the crank end. In this particular case, and when $j\ d$ and $k\ e$ are comparatively

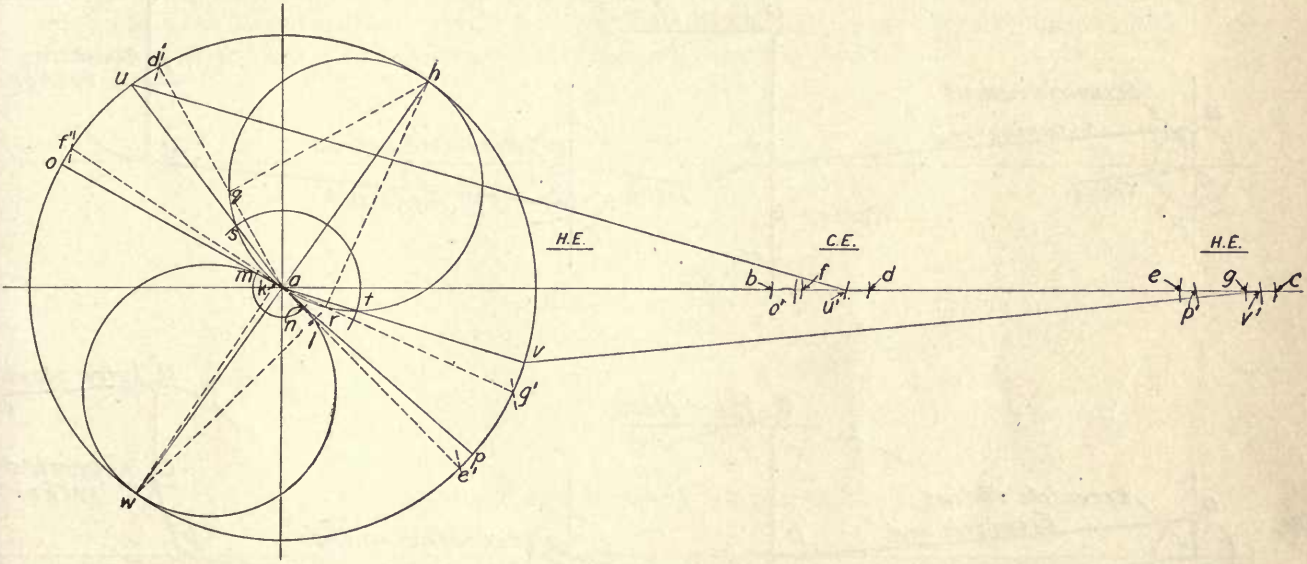


FIG. 22.

small, $o\ f$ and $o\ g$ will be so nearly equal that the difference in values cannot be detected by ordinary graphical construction. Thus both release and compression are equalized on the forward and return strokes, by giving negative exhaust-lap to the head end, and an equal positive exhaust-lap to the crank end of the valve. This irregularity in the construction of the valve, it should be noted, is due to the effect of the varying angularity of the connecting-rod, referred to on a previous page.

General Case.

The above is a special and simple case, and only applies when release and compression both occur at the same percentage of the stroke. In ordinary practice, as a rule, release occurs later than compression, and in such cases equalization of both compression and release are obtained approximately as follows:

In Fig. 22 assume that the valve design has been completed in all respects, except the determination of the exhaust laps. Then the angle of advance, valve-travel, etc., are known.

Assume release on forward stroke (head end card) at f, and

" " " return " (crank end card) at g. $(f\ b = g\ c)$.

Also assume compression on forward stroke (crank end card) at d, and
" " " return " (head end card) " e. $(d\,b = e\,c)$.

The crank-pin positions for the piston positions f, g, d and e, may be found at f' g' d' and e'. Draw the corresponding crank positions shown by the dotted lines. Then the necessary exhaust-lap for release (head end card) at $a\,f'$ is $a\,k$, and the necessary exhaust-lap for compression (head end card) at $a\,e'$ is $a\,l$.

But the exhaust-lap at the head end of the valve cannot have the two different values $a\,k$ and $a\,l$ at the same time. Therefore a compromise is taken by making the head end exhaust-lap $= \frac{1}{2}(a\,k + a\,l) = a\,m = a\,n$.

Drawing crank lines through $a\,m$ and $a\,n$, the corresponding crank-pin positions o and p and pis-

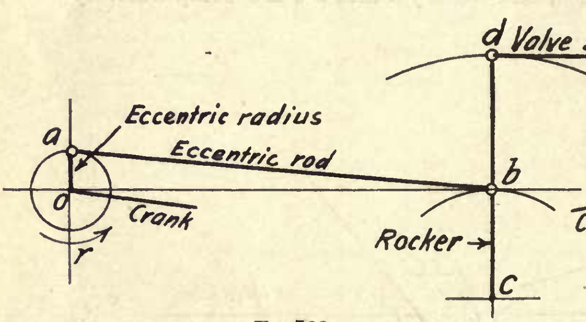


FIG. 23.

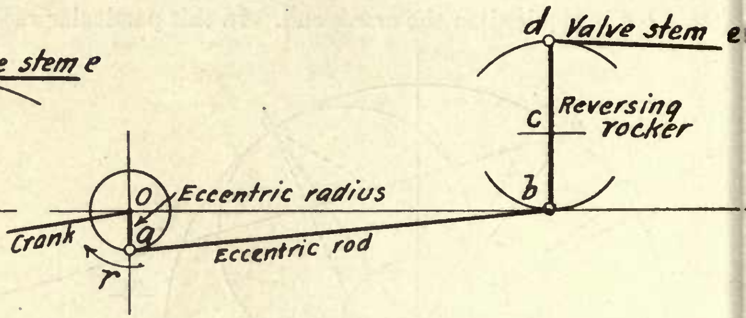


FIG. 25.

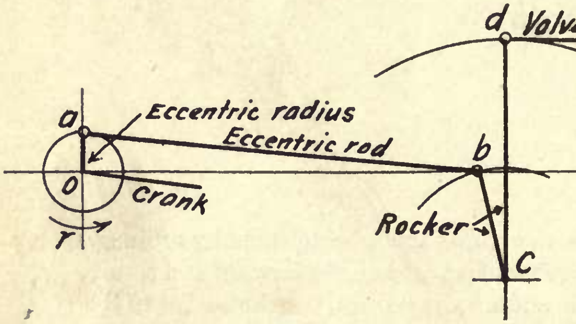


FIG. 24.

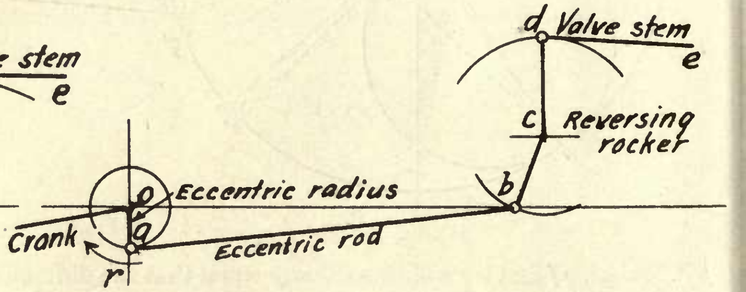


FIG. 26.

ton positions o' and p' may be obtained, o' being release on head end card, and p' compression head end card.

In the same way the compromise lap on the crank end of the valve will be $= \frac{1}{2}(a\,q + a\,r) = a\,s = a\,t$, and release will occur at v' and compression at u'.

$p'\,c$ and $u'\,b$ will now be found to be approximately equal, and the compression on the two ends practically equalized, but not by the same amount as originally laid down at $d\,b$ and $e\,c$. If a definite compression were desired it would have to be found by drawing another trial diagram similar to Fig. 22.

Also the distance $o'\,b$ and $v'\,c$ are approximately equal, and the release on the two ends thus practically equalized, but again not by the same amount as originally laid down at $f\,b$ and $g\,c$.

Rocker-Arms, Straight and Bent, and Their Effect on Valve-Travel and Steam Distribution.

Types of Rocker-Arms.

The principal types of rockers are shown in Figs. 23 to 26. Fig. 23 shows simply a multiplying rocker for accommodating a given location of valve-stem which would otherwise require excessive angularity, or an extremely large eccentric sheave and strap. The rocker in Fig. 24 accomplishes all the above and in addition equalizes cut-off with equal lead when laid out in accordance with the directions on the following pages. The rockers, shown in Figs. 25 and 26, will do all that the ones in Figs. 23 and 24 will do, and, in addition, will produce a different direction of rotation of the shaft, or, in other words, will reverse the direction of running of the engine, as shown by the arrows, *r*.

It was pointed out in the directions for drafting-table, Prob. 1 (p. 17), that the cut-off could be equalized on the two ends of the cylinder by placing unequal steam-laps on the valve, but that this method was objectionable for the reason that it gave very unequal leads.

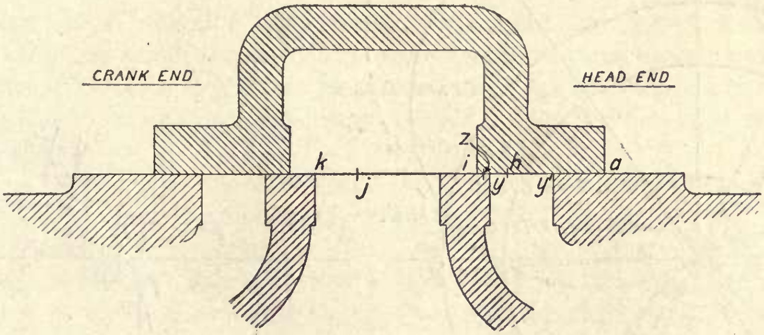


Fig. 27.—Showing valve in central position.

Another method for obtaining equalized cut-off, and at the same time retaining practically equal leads, is by means of the bent rocker. This method permits the use of equal steam-laps on the valve.

Equalizing Cut-Off by a Valve Having Equal Steam-Laps.

In laying out the Zeuner or other valve diagram for a required valve motion, no attention whatever is paid to the rocker-arm. The diagram is always laid out originally as if the eccentric-rod were directly connected to the valve-stem. Allowance for the multiplying action due to unequal lengths of rocker-arms is made in the layout described in the following pages.

The action of the rocker and the effect it has on the motion of the valve may best be shown by a practical application. Keeping in mind the fact that the valve must be the same distance off center when admission begins as it is when cut-off takes place (only going in opposite direction), it may be said in a general way that the rocker is proportioned and situated so as to have the valve in this place at the proper times, despite the effect of the unsymmetrical motion produced by the varying angularity of the connecting-rod. In other words, a bent rocker is a piece of mechanism producing irregular motion, deliberately introduced to counterbalance the irregular motion produced by the connecting-rod. Let Fig. 27 represent the valve and valve-seat.

In Fig. 28, *a d* is the crank position for admission, head end.
a e " " " " " " crank "
a f " " " " " ¾ cut-off, head "
a g " " " " " ¾ " crank "

The circle *d′ f′ e′ g′* is the eccentric circle drawn to the same scale as the crank circle. The angle *d a d′* equals the angle between the crank and eccentric. Therefore,

the eccentric center is at *d′* when admission occurs at *a d*, head end
" " " " " *f′* " cut-off " " *a f* " "
" " " " " *e′* " admission " " *a e* crank end
" " " " " *g′* " cut-off " " *a g* " "

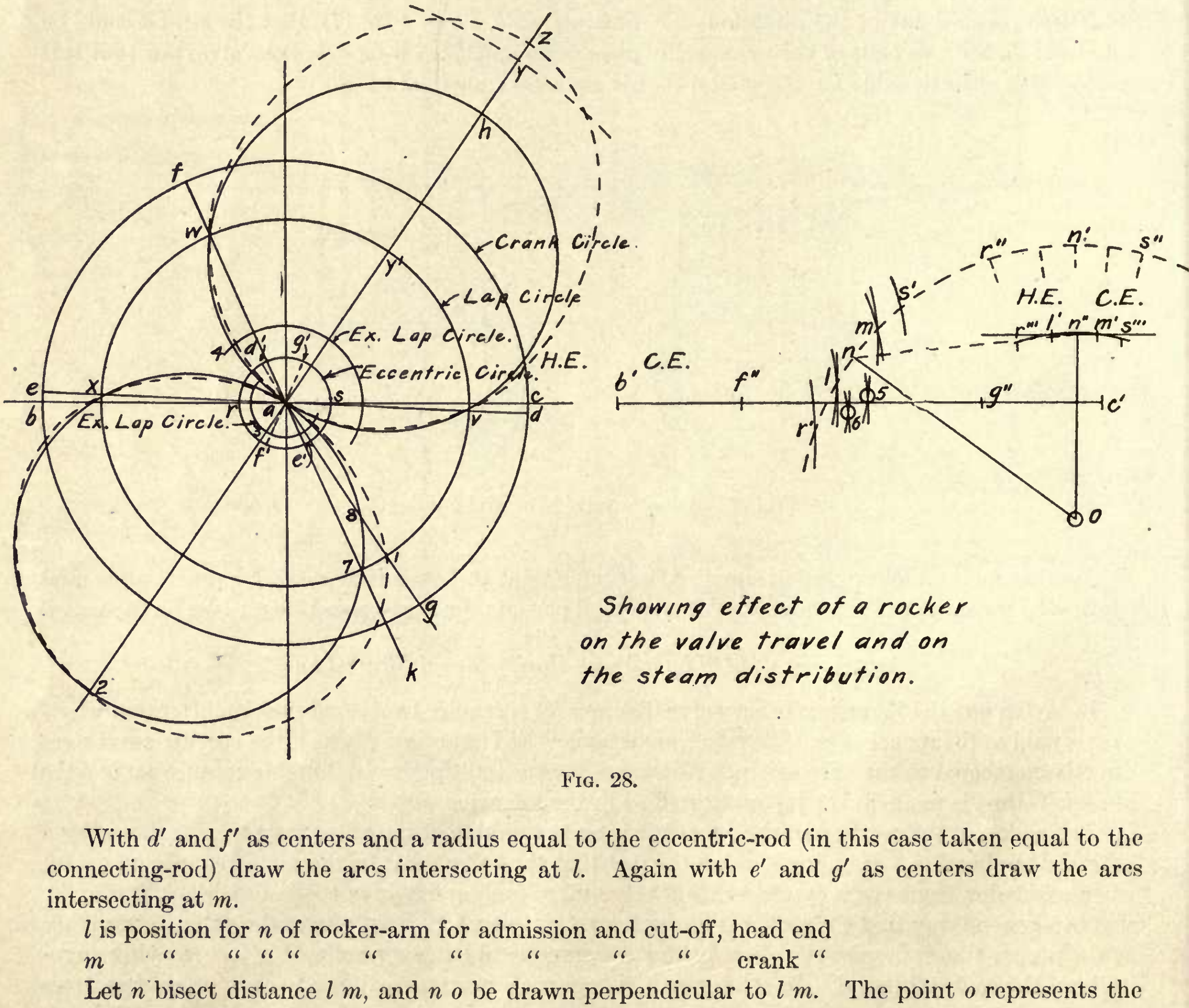


FIG. 28.

With *d′* and *f′* as centers and a radius equal to the eccentric-rod (in this case taken equal to the connecting-rod) draw the arcs intersecting at *l*. Again with *e′* and *g′* as centers draw the arcs intersecting at *m*.

l is position for *n* of rocker-arm for admission and cut-off, head end
m " " " " " " " " " " crank "

Let *n* bisect distance *l m*, and *n o* be drawn perpendicular to *l m*. The point *o* represents the rocker-arm shaft, and its actual location must be such as to afford a convenient attachment of the bearing to the frame of the engine. *o* is assumed here. When the rocker-arm is in the position

$n\ o$ the valve is central. While the end n of the arm is going from n to l the valve must travel to the left, a distance equal to the steam-lap, and the arm $n''\ o$ of the rocker must be proportioned so as to secure this result. This is done by making $n\ o : n''\ o :: n\ l$: steam-lap. n'' will be the end of the valve-stem for central position. The center o for the rock-shaft should be taken so that the arc $l\ m$ when prolonged in both directions will intersect the arcs at r' and s' drawn with r and s as centers, and the length of the eccentric-rod as a radius*. These latter arcs contain the extreme positions of the eccentric-rod, and limit the arc through which the point n oscillates. They also determine the total travel of the valve, and on this account $n\ r'$ and $n\ s'$ should be made as nearly equal as possible, and also as short as is consistent with the necessary valve-travel. This latter may be determined by finding r''', n'' and s'''. Then the horizontal distances between r''' and n'', and between n'' and s''' must at least equal the original half-travel of the valve; otherwise there might be a contracted steam-port opening. o, as already stated, must be a convenient point on the engine frame, and the horizontal line through l' and m' must be at the elevation of the valve-stem. o could be placed on the opposite side of $l\ m$ without affecting the equalization of admission or cut-off.

Unequal Valve Travel on Head and Crank Ends Due to Rocker.

The introduction of the rocker has not only changed the total travel of the valve (compare $r\ s$ with $r'''\ s'''$), but has made the travel from the central position unequal on the head and crank ends. This change in travel may seriously affect some of the other events in the stroke, and must be looked into before the design is considered finished. The effect in this case is shown in Figs. 27 and 28 as follows:

With rocker, the travel of the valve to the left $= n''\ r''' = a\ z$ (on enlarged scale), Fig. 28; equals also $a\ z$, Fig. 27. Without the rocker this travel is $a\ h$ in both Figs. 27 and 28. The increase is therefore $h\ z$, and there is overtravel $= h\ z$. The inside lap of the valve will go to j, Fig. 27 ($i\ j = a\ z$), and the exhaust port-opening will be contracted to $j\ k$, which is less than the width of the steam-port, ($y\ y'$ = width of steam-port) and which will therefore interfere with a free exhaust of steam. In such case the exhaust port must be widened and the valve lengthened to correspond. Such alteration does not interfere with the steam distribution.

Zeuner Circles Changed to Irregular Closed Curves by Rocker.

From the foregoing it is evident that the Zeuner circles used, in designing the valve, do not show the true valve-travel, or the complete true steam distribution, *when* the rocker is added to the valve-gear. To show this, the dotted closed curves $a\ v\ z\ w$ and $a\ 1\ 2\ x$ would have to be drawn. $a\ v = a\ w = a\ 1 = a\ x$ (all on enlarged scale) $= l'\ n'' = n''\ m'$ (both on full size scale). Also $n''\ r''' = a\ z$ and $n''\ s''' = a\ 2$ when brought to the same scale. Intermediate points on the dotted curves may be determined by taking successive positions of the crank and finding the corresponding distances the end of the valve-stem n'' is off center, and setting these distances off on the crank positions. The maximum crank-end travel of the valve, in this case, happens to be the same with the rocker as without.

It will be observed that the effects of the different exhaust-laps which give equalized compression, Fig. 28, are very slightly changed by introducing the bent rocker. This is shown by the dotted curves agreeing very closely with the original Zeuner circles within the limits of the radii of the exhaust-lap arcs, $a\ 3$ and $a\ 4$.

* To be exact the arcs at r' and s' should be the envelopes of a series of arcs drawn with centers on either side of r and s.

In some problems it may be possible to design the rocker so that a symmetrical valve will equalize exhaust and compression in addition to equalizing cut-off and lead, as follows:—Locate point 5, Fig. 28, in the same way that l was found only using the eccentric center positions at release and compression, head end. Also locate point 6 for the crank end. If, in addition to the con-

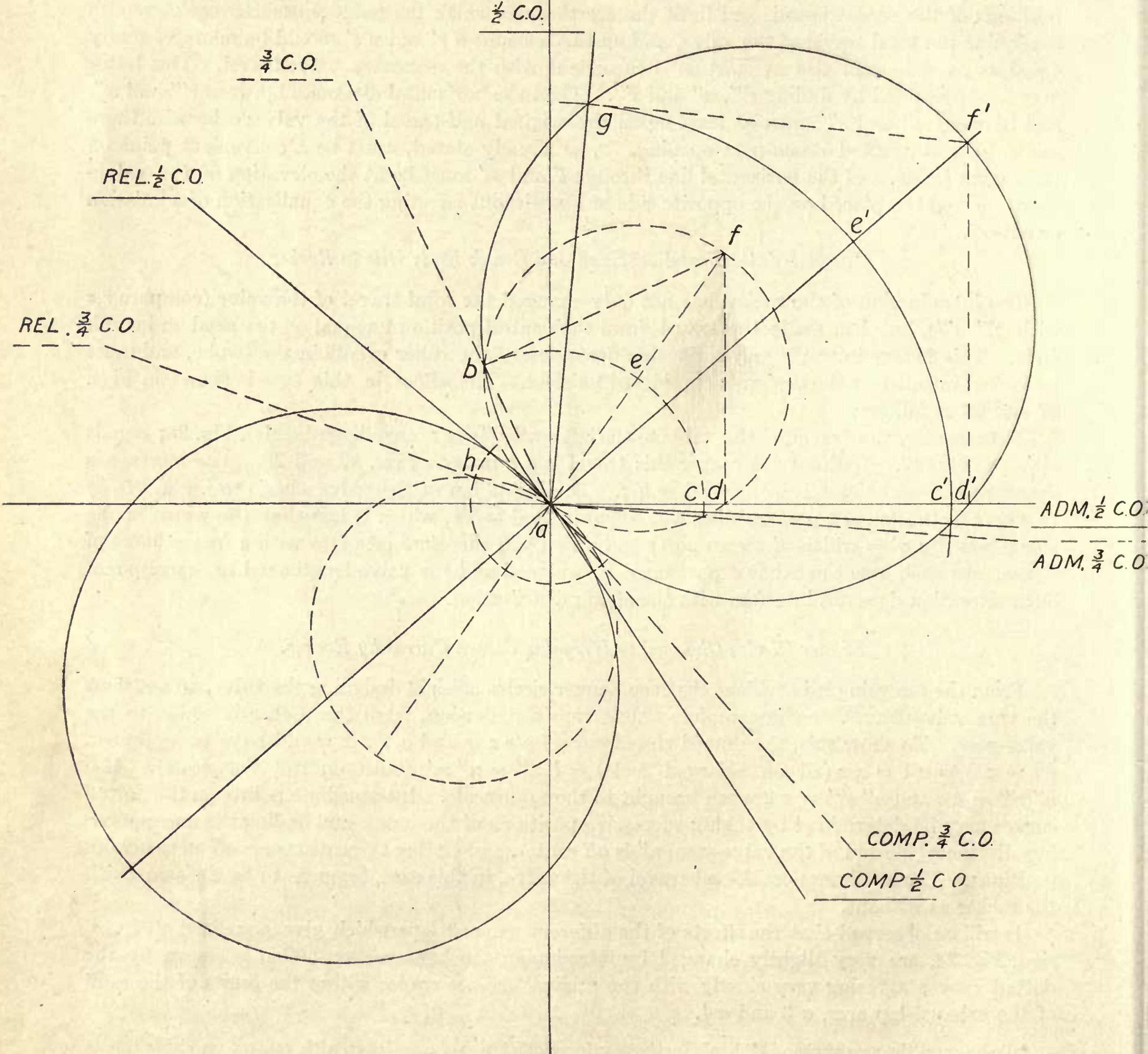


Fig. 29.—Showing Effects of Early Cut-Off with a Plain D Slide-Valve.

siderations already mentioned for determining the position of the rock-shaft o, the arc passing through l and m can be made to include the points 5 and 6, so that they are symmetrical with respect to n, the equalization will be accomplished. With zero inside lap there will be only one point instead of 5 and 6, and this may be made to fall on the arc $l\ m$ at n, and then all four events would be equalized with a symmetrical valve, by the rocker.

Limited Use of the Plain D-Valve.

On page 2 of these directions it was pointed out that a plain slide-valve with zero steam-lap admitted steam to the engine cylinder during the full stroke. Also that to cut off the admission of steam before the end of the stroke, steam-lap had to be added to the valve. This suggests the fact that the earlier the cut-off the larger the steam-lap must be. When steam-lap becomes too large the economical working of the engine is seriously affected, and the plain D-valve, as shown in Fig. 27, can no longer be used economically. The effect of early cut-off is shown in Fig. 29.

Let the dotted construction be the Zeuner diagram for an engine having ¾ cut-off, port-opening $= e\,f$, and lead $= c\,d$. Let the solid construction be the Zeuner diagram for the same engine with ½ cut-off, and with the port-opening and lead the *same as for ¾ cut-off*. Then $e'\,f' = e\,f$ and $c'\,d' = c\,d$.

The lap necessary for ¾ cut-off is $a\ b$. For ½ cut-off it is found to be $a\ g$. The half-valve travel for ¾ cut-off is $a\,f$ and for ½ cut-off it is $a\,f'$. An increase in valve-travel, as may be seen from Fig. 27, calls for a wider exhaust port and therefore a larger valve. A larger valve has additional area exposed to steam pressure and also greater weight of itself, thus giving an increased amount of sliding friction. The additional weight and friction also affect the sensitive action of the governor.

Still further, in Fig. 29, it may be seen that for the earlier cut-off the release and compression are both earlier. (The exhaust-lap $a\ h$ has remained the same for ¾ and ½ cut-off). If the attempt is made to correct the release so as to make it later by increasing the exhaust-lap, the compression is made still earlier.

The above considerations affecting the action of the plain D-valve have established a limit to which, in its simple form, it may be economically used. It is sometimes employed for cut-off at ½, but as a rule not earlier than ⅝ stroke with a fixed eccentric.

Special Valve Exercise.

In addition to drafting-table problem 1, and the several exercises given on pages 11, 12 and 13, the following useful construction by Welch in his treatise on "Valve-Gears" should be noted:

Given: Valve-travel, position of crank for cut-off, and lead. See Fig. 30.

Find: Lap and angle of advance.

$A\ B = $ ½ valve-travel.

$A\ P = $ Position of crank at cut-off.

With A as a center and $A\ B$ as a radius, draw a circle intersecting the horizontal line $A\ J$ at D. With D as a center and a radius equal the lead, draw the "lead-circle" $T\ H\ J$.

From P draw line $P\ K$ tangent to the lead-circle.

Draw $A\ K$. Draw the circle $V\ Z\ R$ tangent to $P\ K$.

Then $V\ Z\ R$ = lap circle. Draw $A\ E \perp$ to $P\ K$.

Then $\angle B\ A\ E$ = Angle of advance.

$A\ K$ = Position of crank at admission and $Q\ L = D\ T$ = lead.

Proof That $Q\ L = D\ T$.

Draw $E\ R$ and $E\ L \perp$ to $A\ K$ and $A\ D$ respectively.

Draw $D\ F$ parallel to $K\ Z$. Draw $D\ H$ parallel and equal to $F\ Z$.

In $\triangle$ $E\ A\ R$ and $K\ A\ Z$; $E\ A = A\ K$, $\angle E\ A\ K$ is common, and $\angle$ $E\ R\ A$ and $K\ Z\ A$ are right $\angle$.

$\therefore \triangle s\ E\,A\,R$ and $K\,A\,Z$ are equal, and $A\,Z = A\,R$. Also A Q $= A\,R = A\,Z$ by construction.
In $\triangle s\ E\,A\,L$ and $D\,A\,F$; $E\,A = A\,D$, $< E\,A\,D$ is common, and $\angle s\ E\,L\,A$ and $D\,F\,A$ are right $\angle s$.
$\therefore \triangle s\ E\,A\,L$ and $D\,A\,F$ are equal and $F\,A = A\,L$.
$\therefore Q\,L = F\,Z$. But $F\,Z = D\,H = T\,D$ by construction.
$\therefore Q\,L = T\,D =$ lead.

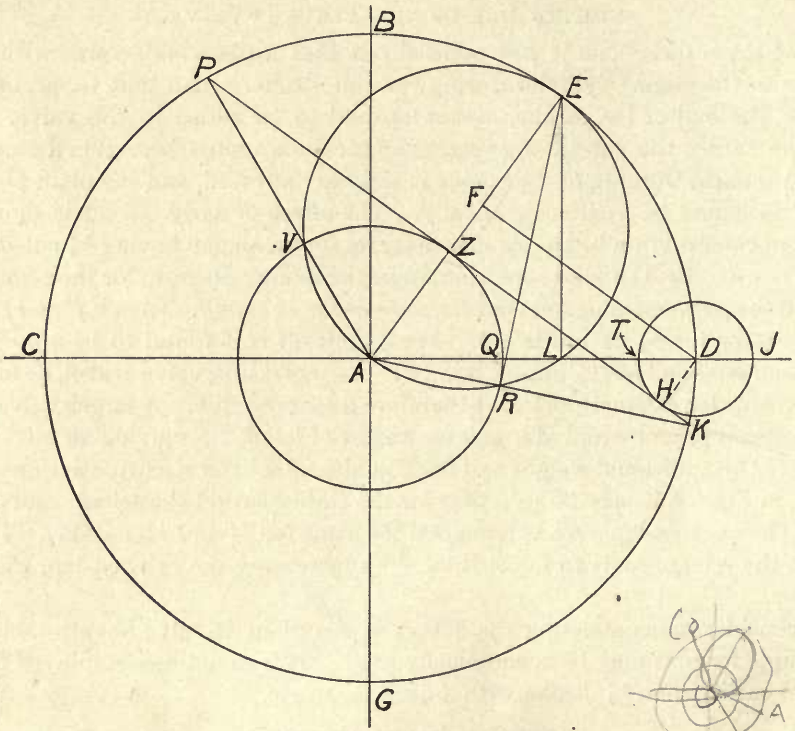


Fig. 30

The Allen Valve.

This valve is the type common in locomotive work. With a given travel it admits twice as much steam as the ordinary valve. For example, in Fig. 31, the valve is in its extreme right-hand position, having moved a distance equal to the lap + ½ the port-opening, $b\,c$. With this half-travel the port-opening is $(b\,c + d\,e)$ or $2\,b\,c$. The steam-lap $= k\,c - h\,b$ which should be equal to, or greater than $c\,e$ in order to keep the two ends of the cylinder from being in communication through the passageway in the valve. The edges of the valve-seat, a and m, must have special consideration in this type of valve, namely, that r on the valve must pass m on the valve-seat exactly at the same time that c on the other end of the valve is passing b. A top view of the Allen valve, or "trick valve" as it is sometimes called, with so much of the valve-seat as is visible, is shown on reduced scale in Fig. 31a.

Drafting Table Problem, No. II. Design for an Allen Valve.

Let it be required to design an Allen valve for an engine having a. . . .inch bore,. . . .inch stroke, running at. . . .revolutions per minute and cutting off at. . . .stroke on both ends, with. . . .inch lead on head end. Use zero exhaust-lap on both ends. The steam-port may be taken .7 of the bore. The connecting-rod may be taken 5 times the crank length.

The first step in this design consists, as in problem I, in calculating the width of port-opening, and width of steam-port. Then lay out the crank circle and cross-head travel to the largest regular scale that the drawing paper will allow. Judgment must be used in determining the scale for the Zeuner diagram, based on the principle that the largest convenient scale for geometrical drawings, gives the greatest accuracy.

Effect of Two Admission- and Two Lead-Areas on Zeuner Diagram Construction.

The ½ valve-travel in drafting-table problem, I, was equal to the lap + the whole width of the steam-port opening. In this problem it is equal to the lap + ½ the calculated width of the steam-port opening, because the Allen valve is in effect two plain D-valves combined, and admits steam to the same steam-port through two openings at the same time. Each opening then takes care of ½ the lead, and in laying out the Zeuner diagram for the Allen valve only ½ the given lead as well as ½ the calculated width of the steam-port opening is considered. When the Zeuner diagram is completed, designate the crank positions in the manner shown in Fig. 29.

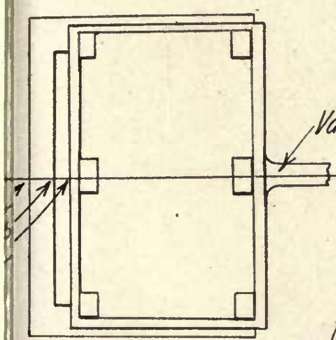


FIG. 31a.—Top view of Allen valve and valve seat.

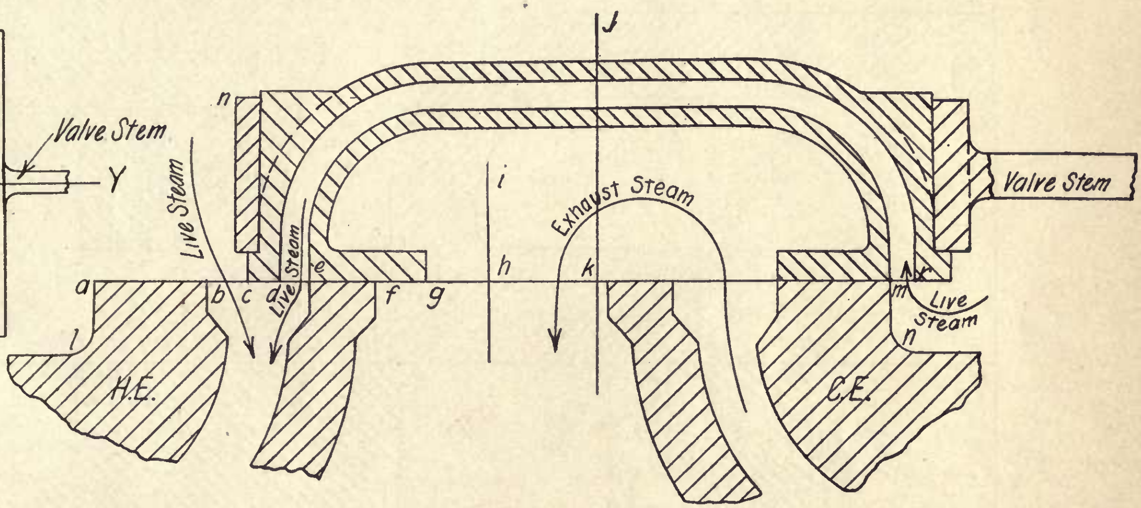


FIG. 31.—Allen Valve. Section on $x\ y$ of FIG. 31a.

The valve may now be laid out full size. Starting with the point b, Fig. 32, which is an illustration for a general case, $b\ e$ = steam-lap. $b\ c$ = thickness of flange of the outside valve wall, which should be a little wider than the valve wall to allow for facing, and small enough so that $b\ c + c\ d$ is equal to or less than the steam-lap: otherwise the two ends of the cylinder might be in communication through the auxiliary passage X for an instant. If $c\ b$ should come so small as to prevent a good casting, some one or more conditions of the design would have to be changed.

$c\ d$ = ½ width of calculated steam-port opening.

$e\ f$* = $2\ c\ d + b\ c$.

$f\ g$ = exhaust-lap.

$f\ h$ = width of bridge = ⅝″ for engine given in this problem.

$h\ i$ = ½ width of exhaust port.

* When $e\,f$ is much larger than the calculated width of port, it may be reduced to the correct size in some such way as shown by the dotted lines at k and l (= calculated width of port) in Fig. 32; or, a face-plate may be used. These dotted lines are not to be drawn by the student.

Total width of exhaust port = maximum exhaust-lap + ½ travel of valve + width of steam-port − width of bridge.

The location of the point *a* is important, for just as the point corresponding to *b* on the other end of the valve is uncovering the point corresponding to *e*, the point *c* must be uncovering *a*. Therefore *c a* must equal the steam-lap on the other end of the valve.

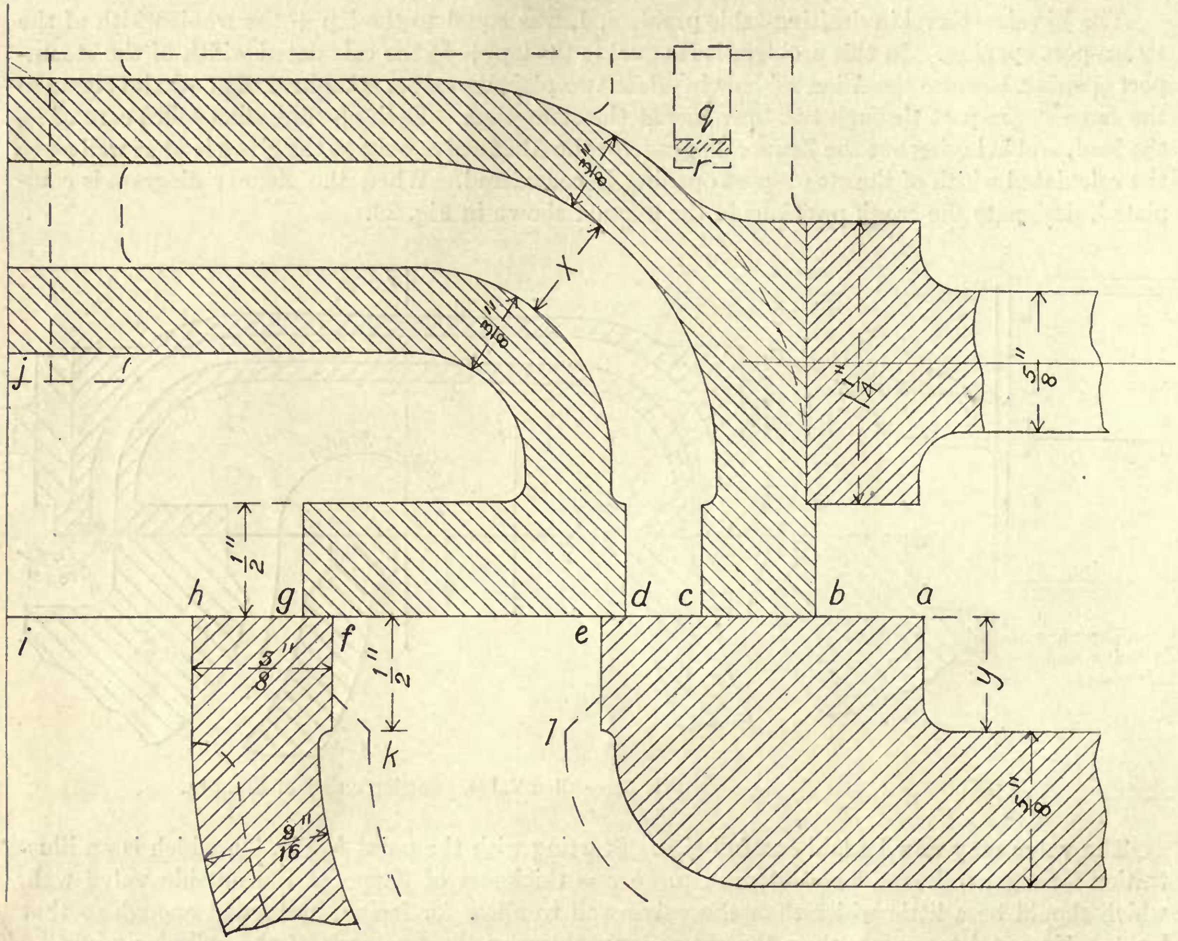


Fig. 32.

It will be evident that *y* must be at least equal to *c d* so as to give free admission to the valve passage *X*. It may be made equal to *c d* + ⅛″.

X may also be made equal to *c d* + ⅛″ to allow for friction of steam in the rough cored passage-way.

i j may be taken = ½ (calculated width of steam-port + width of exhaust-port.)

All the remaining dimensions necessary to complete the design are independent of the action of the valve so far as steam distribution is concerned, and are determined solely according to the size of the valve. For this problem they may be taken as shown in Fig. 32.

Locomotive Balanced Valve.

The Allen valve may be converted into the ordinary locomotive balanced valve by adding metal in the form of the dotted lines shown at the top of the valve, which the student should do. *q* is a slot containing a rectangular bar which is kept against the faced surface of the cover of the steam-chest, as the valve slides back and forth, by means of a spring at *r*. This slot and the bars extend all around the valve, and keep the live-steam pressure from exerting its power to press on the back or top of the valve and thus force it against the valve-seat with greatly increased friction. In the locomotive balanced valve there is a small hole through the center communicating with the exhaust. This carries off any live steam that may leak through.

Place the necessary working dimensions and mark the finished surfaces on the design.

Tabulate the results as in drafting-table problem, No. 1.

Limited Use of the Allen Valve.

That the Allen valve cannot be used to advantage for cut-off much later than half stroke with a fixed eccentric may be seen by referring to Figs. 33–36. A valve is shown in part section

Fig. 33

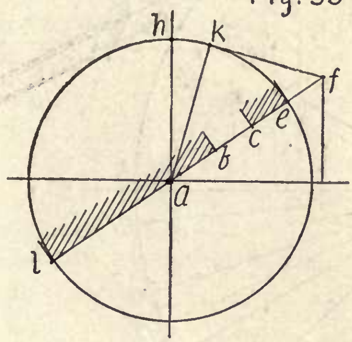
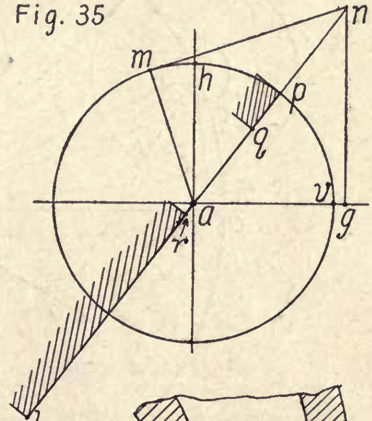


Fig. 35



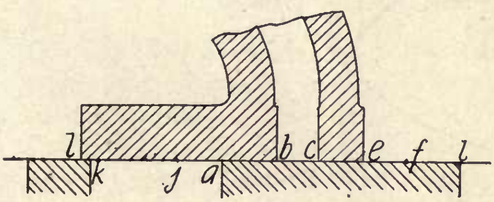


Fig. 34.

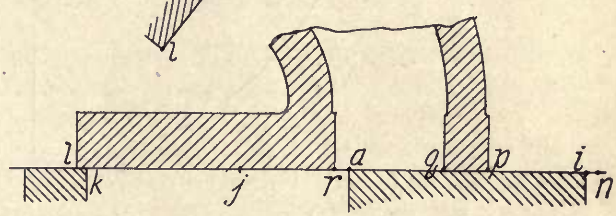


Fig. 36

in Fig. 34 and its corresponding Zeuner diagram is shown in Fig. 33, both illustrations being to the same scale and similarly lettered. It will be noted that the lap, *a e*, is large enough to contain the valve-wall thickness, *c e*, the passageway, *b c*, equal to *e f*, and to have some space, *b a*, left over. But in Fig. 35, let the cut-off be assumed at *a m*, keeping the same lead and lap; then the single port opening becomes *p n*. If now the necessary valve-wall thickness is laid off equal to say *p q*, it is found that there is not enough room in the steam-lap for another passageway equal to *p n*, and therefore that cut-off as late as *a m* is not possible with an Allen valve having the data here used and direct-connected by a "fixed" eccentric. With special forms of valve gears and governors, the travel of the valve and the angle of advance may be varied automatically, and the cut-off made later, as will be shown when the different forms of valve gears and governors are taken up.

SECTION II.—VALVE DIAGRAMS.

In addition to the Zeuner diagram a number of methods have been devised to show graphically the relative positions of the valve and crank at any instant. A brief description of the more important diagrams will be given.

Bilgram Diagram.

Let it be considered that the crank is on the head end dead-center on the line *a b*, Fig 37, and turning in the direction shown by the arrow. Also let the distance *a b* represent the ½ valve-

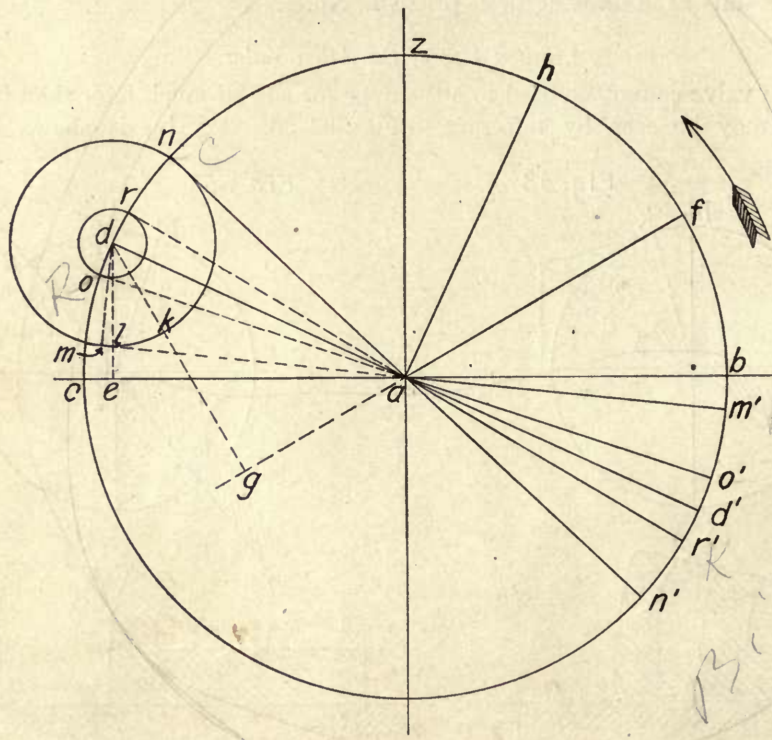


Fig. 37.

travel, and draw the eccentric center circle *h b c*. Then on the opposite side of the crank lay off the angle *c a d* = the angle of advance.

From *d* draw the line *d e* perpendicular to the crank position *b a* prolonged, and *d e* will be the distance the valve is off center when the crank is at *a b*. Likewise,

when the crank is at *a f* the valve is off center *d g*, and
" " " " " *a h* " " " " " *d a* (maximum distance)
" " " " " *a d* " " " central.

The reason for these facts may be found in Fig. 38, where $a'f'$ is an assumed position of the crank, and $a'\,z$ the corresponding eccentric position, with $x\,a'\,z$ (= $c\,a\,d$, Fig. 27) as the angle of advance. Then with the crank at $a'\,f'$ the valve is off center a distance $z\,y$. It remains to show that $z\,y = d'\,g'$.

$< x a' z = < d' a' c'$ = angle of advance. Since $x a'$ is $\perp$ to $a' w$, and $v a' \perp$ to $a' c'$, the $< x a' v = < c' a' w$. Therefore $< d' a' w = < z a' v$. Also $\angle$s $d' g' a'$ and $z y a'$ are right angles, and the $\triangle$s $d' a' g'$ and $z a' y$ are equal. $\therefore y z = d' g'$.

Since $d g$, Fig. 37, equals the distance the valve is off center, it is only necessary to draw a circle with $d k$ (= the steam-lap) as a radius and $k g$ will equal the steam-port opening for the crank position $a f$.

$l e$ = the lead. $a m'$, drawn with its prolongation tangent to the steam-lap circle, is the position of the crank for admission. $a n$, also tangent to the steam-lap circle, is the cut-off position.

If the valve has an exhaust-lap $d o$, the exhaust-lap circle should be drawn with a radius = $d o$, and then $a o$, tangent to it, will be the crank position for release. Likewise $a r'$, with its prolongation tangent to the exhaust-lap circle, will be the position for compression.

For the crank end of the cylinder $a m$ is admission; $a n'$ cut-off; $a o'$ release, and $a r$ compression. In Fig. 37 the steam and exhaust-lap circles are taken the same on both the head and crank ends.

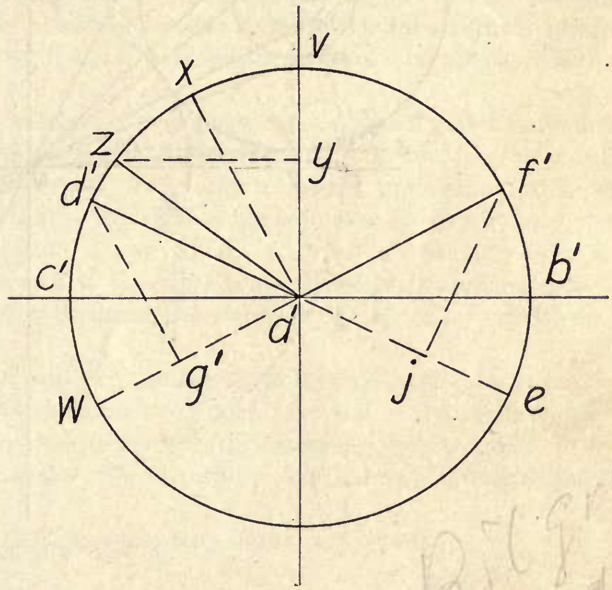


FIG. 38.

If the head end exhaust-lap had been negative, $a r$ instead of $a o$ would have been the release position for the head end.

Solution of Drafting Table Problem No. 1 by Bilgram Diagram.

Drafting table problem 1, of this course, would be solved by the Bilgram diagram as follows:

Given: Cut-off, release, lead, and steam-port opening. To find: Steam and exhaust-laps, travel of valve, and crank positions for the events of the stroke.

In Fig. 39 draw line $c b$ for center-line of engine. At any point a draw $a n$ for the given cut-off position. Draw line $s t$ parallel to $c b$ and a distance from it equal to the lead. Draw arc $u v$ about a as a center with the calculated width of steam-port opening as a radius. Find by trial the center, d, of a circle that will be tangent to $a n$, $s t$, and the arc $u v$.

Then $d l$ is the required steam-lap. Draw $a o$ for the given release position, and determine the exhaust-lap by drawing a circle with d as a center and tangent to $a o$.

If $d w$ represents the necessary exhaust-lap circle for the crank end to give equalized compression, as called for in drafting-table problem 1, then the events of the stroke are as follows:

	Admission.	Cut-Off.	Release.	Compression.
Head end	$a m$	$a n$	$a o$	$a g$
Crank end	$a q$	$a x$	$a y$	$a z$

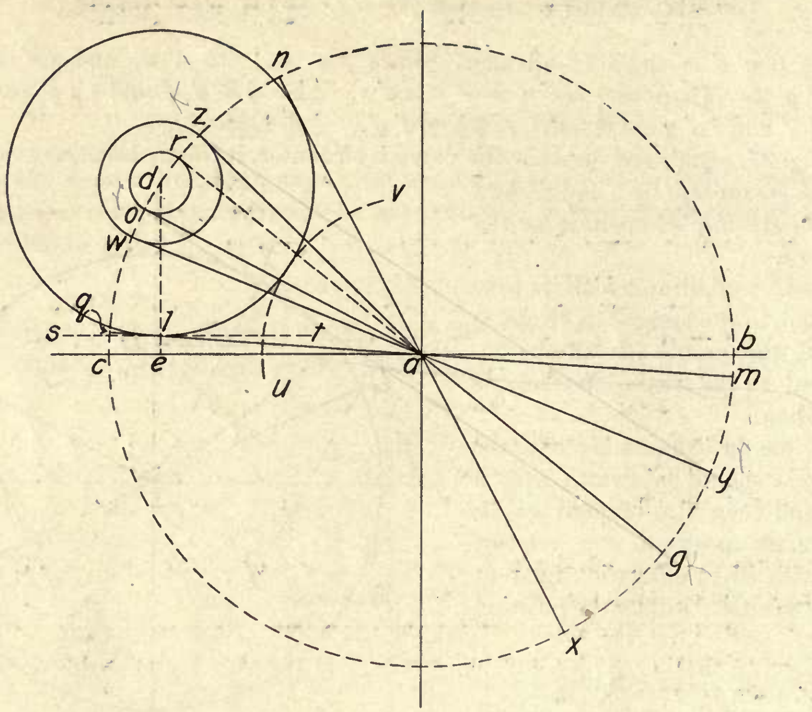


FIG. 39.

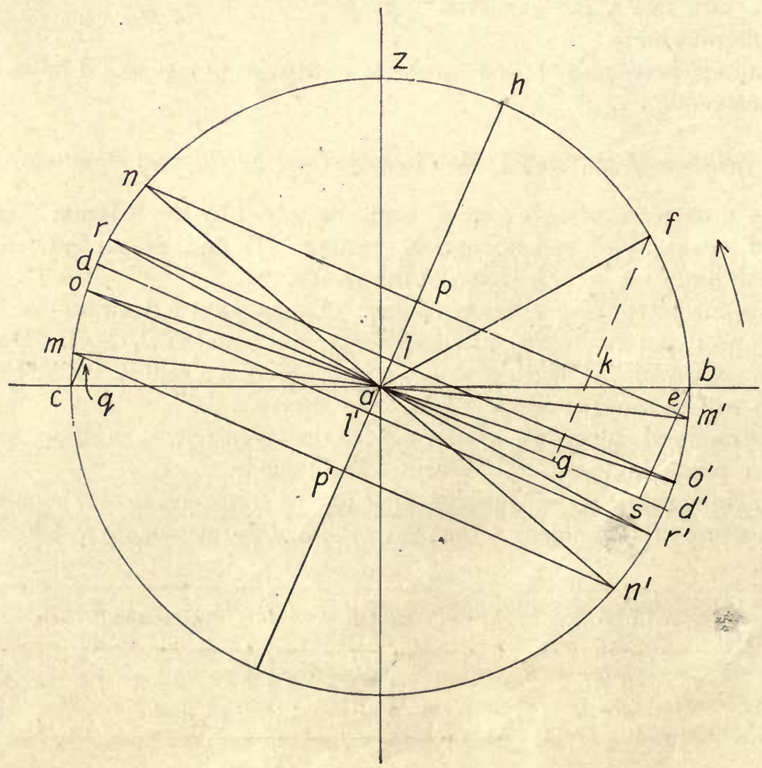


FIG. 40.

Reuleaux Diagram.

In making use of this diagram let the indefinite line $c\ a\ b$, Fig. 40, be the center-line of the engine, and at any point, a, draw $a\ z$ perpendicular to it. Draw the circle $c\ z\ b$ with a radius equal to the ½ travel of the valve, and lay out the angles $z\ a\ h$ and $b\ a\ d'$ each equal to the angle of advance. Then for any position of the crank such as $a\ f$, it is only necessary to draw from f, the point where the crank line crosses the valve circle a, perpendicular to the line $a\ d'$ limiting the angle of advance, to find the distance $f\ g$ that the valve is off center.

This may be proven by reference to Fig. 38, where $a'\ f'$ is a given crank position and $d'\ e$ the line limiting the angle of advance ($= b'\ a'\ e$). Since z is the eccentric-center position for the crank position $a'\ f'$, $z\ y$ is the distance the valve is off center. It is only necessary to demonstrate that $f'\ j = z\ y$. This is readily done for the reason that in the right-angle triangles, $z\ a'\ y$ and $f'\ a'\ j$, the sides $a'\ z$ and $a'\ f'$ are equal, and the angles $z\ a'\ y$ and $f'\ a'\ j$ are equal. (Angle $b'\ a'\ e$ = angle $z\ a'\ x$ = angle of advance; and angle $b'\ a'\ f'$ = angle $x\ a'\ v$, the sides being respectively perpendicular). Therefore the triangles are equal, and $f'\ j = z\ y$ = distance the valve is off center for crank position $a'\ f'$.

In Fig. 40 draw the line $n\ m'$ parallel to $d\ a\ d'$ and at a distance from it equal to the steam-lap = $a\ p$. Then for any crank position such as $a\ f$ the steam-port is open $f\ k$. $a\ p$ = the steam-lap for one end of the valve, and $a\ p'$ the steam-lap for the other end. In this case they are equal.

Likewise $a\ l$ = the exhaust-lap for one end, and $a\ l'$ for the other. Lines drawn through l and l' parallel to $d\ d'$ are the exhaust-lap lines.

The events of the stroke according to the Reuleaux diagram, occur as follows:

	Admission.	Cut-Off.	Release.	Compression.
Head end	$a\ m'$	$a\ n$	$a\ o$	$a\ r'$
Crank end	$a\ m$	$a\ n'$	$a\ o'$	$a\ r$

$b\ e$ is the lead for the head end, and $c\ q$ for the crank end.

Valve Ellipse.

The valve ellipse is a curve in which the ordinates show the amount the valve is off center; and the abscissæ, the corresponding piston positions.

It may be obtained, as in Fig. 41, by dividing the cross-head travel into an equal number of parts as at 1, 2, 3, etc. With these divisions as centers, and with a radius equal to the length of the connecting-rod, strike arcs intersecting the crank-pin circle in the points 1′, 2′, 3′, etc.

$a\ r$ is the radius of the eccentric-center circle, and the angle $o'\ a\ r$ is the angle between the crank and the eccentric. The points $r\ e\ f\ g$, etc., show the eccentric-center positions for the corresponding crank-pin positions 0′, 1′, 2′, 3′, etc.

Then with piston at 0 the valve is off center the distance $r\ h = o'\ l$
" " " 1 " " " " " " " $e\ i = d\ m$
" " " 2 " " " " " " " $f\ j = c\ n$

etc., and these values, $o'\ l$, $d\ m$, $c\ n$, etc., laid off on the piston position ordinates through o', d, c, etc., in the valve ellipse diagram determine the points on the valve ellipse. The points d, c, b, etc., are equally spaced the same as the points, 1, 2, 3, etc., in the cross-head stroke.

The curve generated in this way, although called the "valve ellipse," is not a true ellipse, unless the connecting-rod is of infinite length in which case the points 1′, 2′, 3′, etc., would lie on the ordi-

nates through *d, c, b,* etc. In Fig. 42 the dotted curve is for the mechanical equivalent of the infinite connecting-rod, and is a true ellipse, while the full curve is for a connecting-rod = 4 crank lengths.

For the crank position *a d,* or the equivalent piston position *e h,* Fig. 42, and the finite connecting-rod, the valve is off center the distance *h e.* If the valve has a lap = *h f* the steam-port opening for this position is *f e* on the head end, and if the exhaust-lap on the crank end of the valve is *h g* the opening of the port to exhaust is *g e.* The steam and exhaust-lap lines are drawn parallel to the center-line *b c.*

It follows, then, that the point in which the steam-lap line intersects the valve ellipse determines directly the piston positions, and indirectly the crank positions at admission, cut-off, etc.

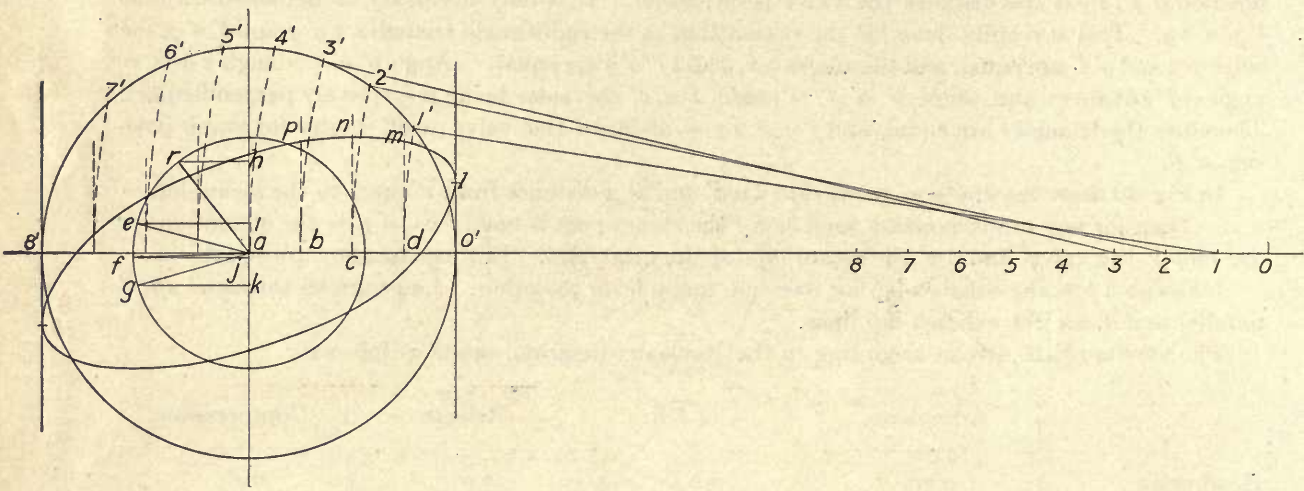


FIG. 41.

For cut-off on the head end, finite card, the piston is at l'', directly below *l,* Fig. 42. Projecting *l* to the center-line and drawing an arc with the connecting-rod as the radius, *a m* is obtained for the crank position at cut-off. In like manner the crank positions for release, compression and admission may be found.

Method of Determining Steam and Exhaust-Port Openings, and Steam and Exhaust-Laps by Combining the Valve Ellipse and Indicator Cards, and Without Removing Steam Chest Cover.

By combining the valve ellipse and the indicator card, as shown at Figs. 42 and 43, a ready means is afforded for examining the steam distribution of a plain D- or piston-valve, and also for determining the steam and exhaust laps, *without removing the steam chest cover or disturbing the valve or the running of the engine in any way.* This is done by taking the indicator card from the engine in the usual way, and at the same time taking the valve ellipse automatically from the engine, and drawing the two curves one under the other as in Figs. 42 and 43. The full-line valve ellipse is, so far then, all that is known in Fig. 42. In obtaining the valve ellipse the abscissæ displacements would come from the cross-head, and the ordinate displacements from the valve stem. To get at the information which it contains, first draw the lines *s t* and *u v* tangent to the ellipse and parallel to the stroke line, which would be recorded in obtaining the ellipse card. The perpendicular distance betewen *s t* and *u v* is the valve-travel. Draw the line *b c* midway between *s t* and *u v.*

On the indicator card determine the points l' (cut-off), n' (release), p' (compression) and q' (admission), and project these points up to the ellipse at l, n, p, and q. Then since admission and cut-off are both governed by the steam-lap, the points l and q should each be a distance from $b\ c$ equal to the steam-lap, and lie on the steam-lap line, which is thus determined. Likewise n and p should each be the same distance from $b\ c$, and determine the exhaust-lap.

The laps of the valve, together with its travel, and also the steam and exhaust-port opening at any instant, are now known for the head end. For the crank end, the crank end indicator card and ellipse would be taken in the same manner.

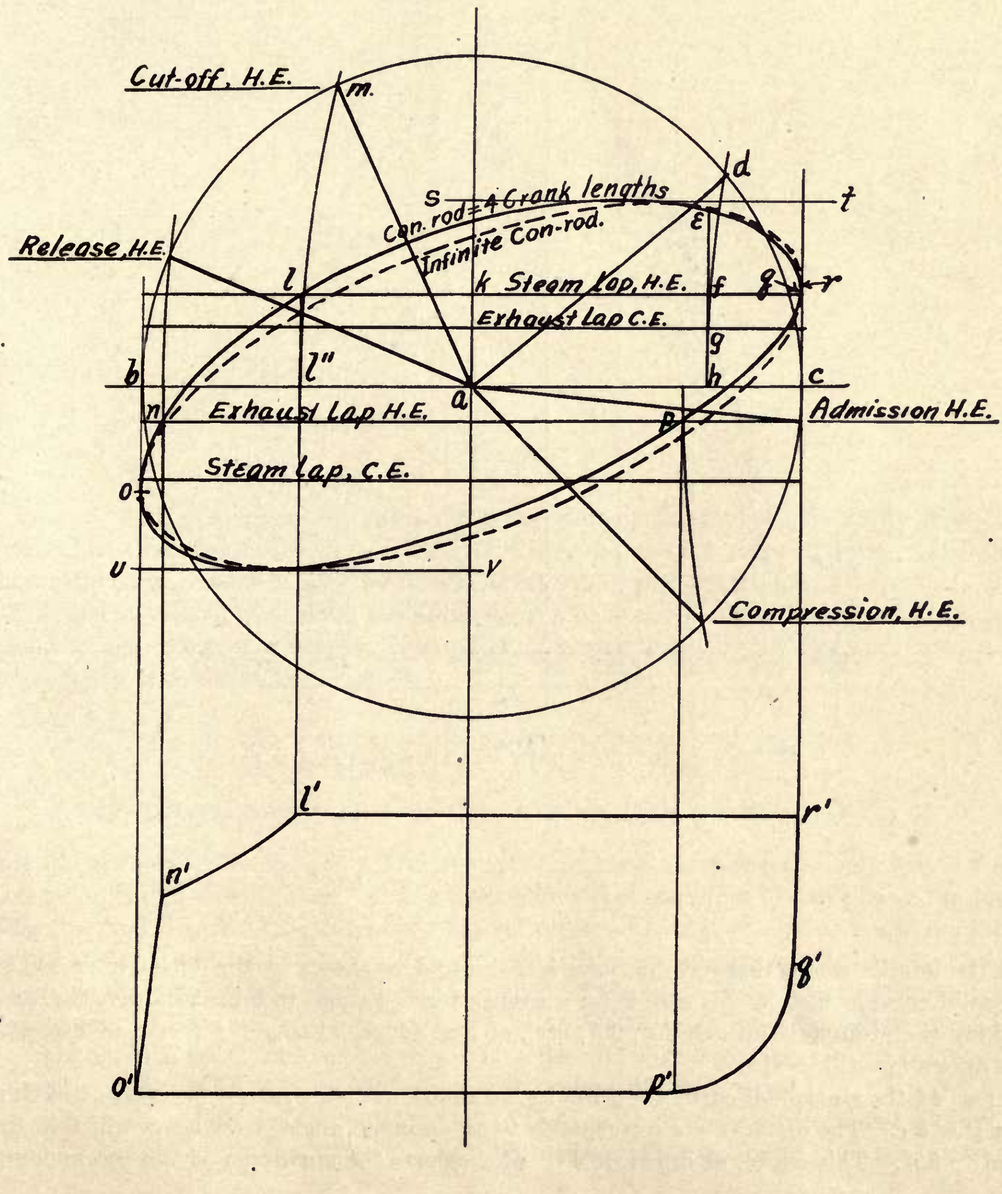


FIG. 42.

FIG. 43.

Sinusoidal Diagram.

In this diagram the crank angles are laid off on the abscissa line, and the piston and valve displacements on the ordinates.

The sinusoidal diagram affords a ready means for studying the steam distribution for setting the eccentric so as to secure the best results for a given engine.

In Fig. 44 take a distance such as $b\ c$ to represent 360 degrees, and divide the line into a convenient number of equal parts. On the ordinates through the division points lay off distances equal to the corresponding piston displacements, thus obtaining a curve through the points $c\ d\ b$.

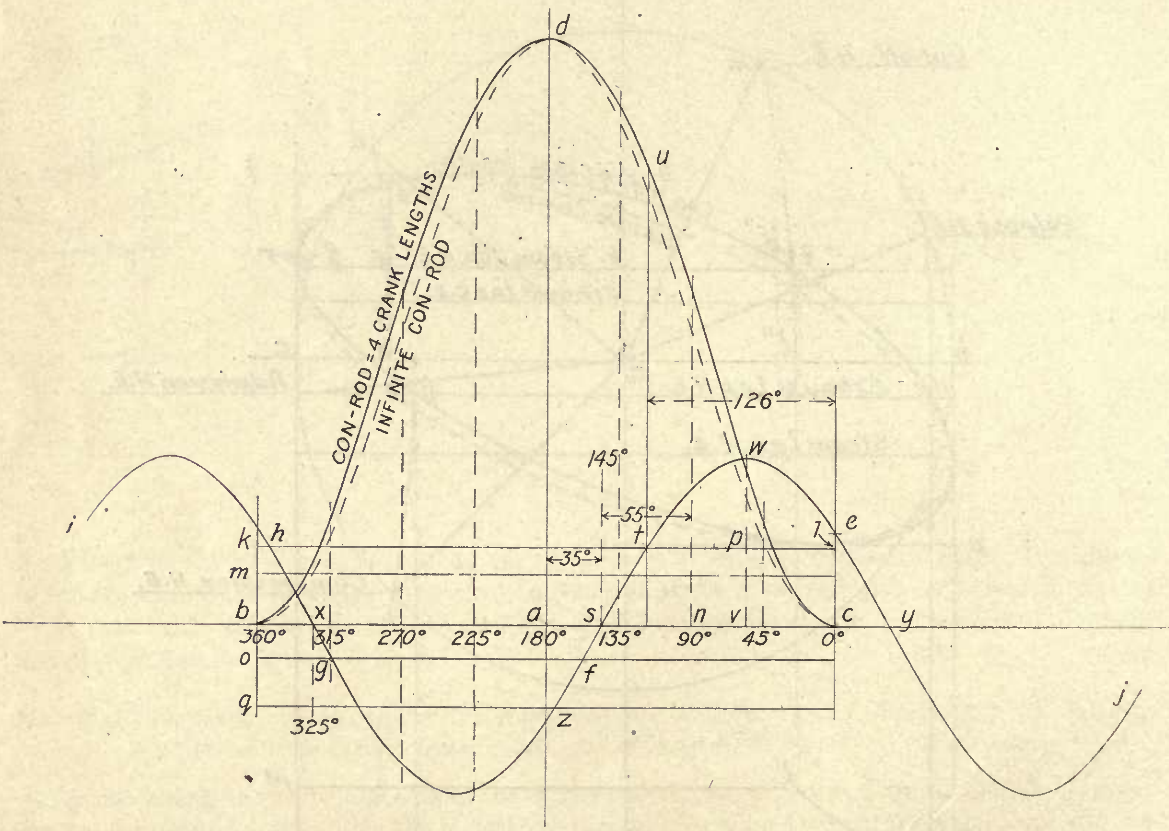


Fig. 44.

For the infinite connecting-rod the curve is a sinusoid, as shown by the dotted line. The points on the solid curve, which in this case is for a connecting-rod equal to four crank lengths, are found by making the ordinates through 45°, 90°, etc., of Fig. 44, equal to $g\ d$, $f\ d$, etc., of Fig. 45, which is here drawn two-thirds size.

In Fig. 44 the sinusoidal curve $i\ w\ j$ has its maximum ordinate $v\ w = \frac{1}{2}$ valve-travel, and its pitch $x\ y = b\ c$. The distance $a\ s$ corresponds to an assumed angle of advance, which in this case is equal to 35°. This curve, as drawn in Fig. 44, neglects the angularity of the eccentric-rod, and

is a sinusoid. It may be so used practically, unless an investigation is one of much precision, in which case the actual valve curve may be found by the method shown in Fig. 45.

In Fig. 44, let $b\ k$ = the steam-lap, head end.

$b\ q$ = " " " crank end.

$b\ o$ = " exhaust-lap, head end.

$b\ m$ = " " " crank end.

Draw lines through k, q, o and m, parallel to $b\ c$. Then cut-off, head end, occurs at t (126°), release at f (158°), compression at g (315°), and admission at h (355°). The degrees here given are for illustration, and therefore only approximately correct. $l\ e$ is the lead.

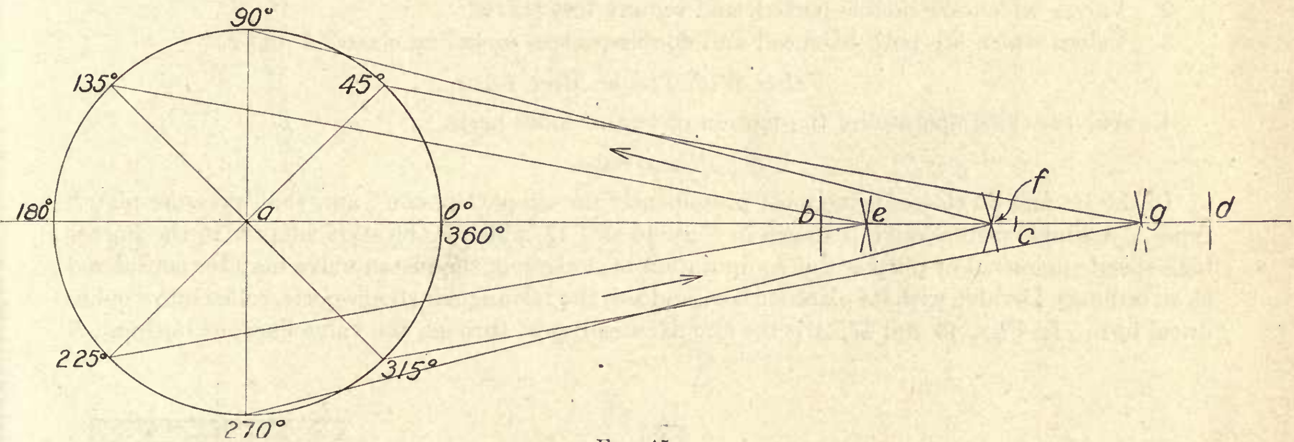


FIG. 45.

In order to use this diagram for determining the effect of different angles of advance, the valve curve should be extended as shown to i and j, and then drawn on a piece of tracing paper or cloth. By placing the curve so that s falls on a the events of the stroke for zero angle of advance are found at once; with s at n (90°) the events for 90° angle of advance are known. For intermediate angles of advance s falls between a and n. The effect of changing the valve laps may also be readily shown by raising or lowering the lap lines.

SECTION III.—TYPES OF VALVES.

EFFECT OF FRICTION DUE TO PRESSURE ON BACK OF PLAIN D-VALVE.

Thus far in these notes the plain D-valve and the Allen valve are the only types that have been treated in the classroom work. It has been shown that the plain D-valve has a limited range of application, and that for cut-offs earlier than ⅝ stroke a D-valve of excessive and impracticable weight and travel would be required. Furthermore, the steam-pressure on the back of the plain D-valve, in addition to the pressure due to the weight of the valve (when in a horizontal position), increases largely the friction on the valve-seat. In an engine of 14-in. bore, with a plain D-valve 8½ ins. long and 12½ ins. wide, and a steam pressure of 70 lbs., the pressure on the back of the valve would be $8.5 \times 12.5 \times 70 = 7437$ lbs., or 3.7 tons. When a valve is so designed that this pressure cannot act on any considerable portion of it so as to produce pressure and friction on the valve-seat, it is said to be "balanced." In engines (especially those of high speed) where the

adjustment of the cut-off, etc., is produced by the effort of the governor in changing the position of the valve, the friction due to such high pressure is too great to be properly overcome directly.

Classification of Valves.

Other forms of valves must, therefore, be adopted, and these forms may be roughly classed as follows, an exact classification and a simple one that would include the numerous variations and combinations of types being practically impossible:

One-Piece Valves.

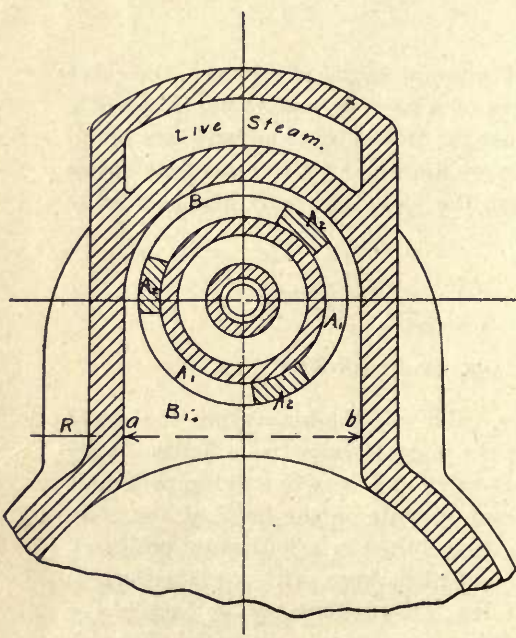
1. Valves in which the steam pressure is balanced, permitting large travel if desired.
2. Valves which are double-ported, and require less travel.
3. Valves which are both balanced and double-ported, including classes 1 and 2.

Valves With Two or More Parts.

4. Valves which operate by the motion of two or more parts.

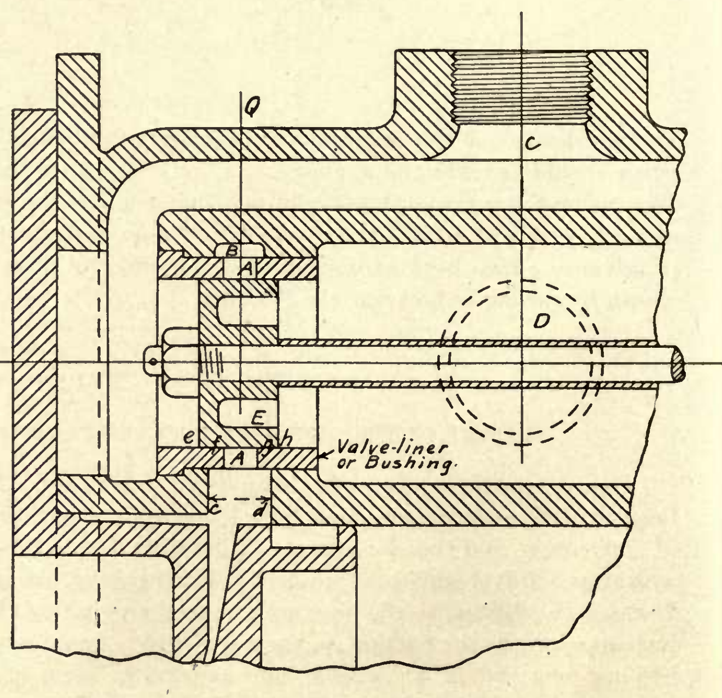
Piston-Valve.

In the 1st and 3d classes come most prominently the simple "piston" and the "pressure-plate" types. A simple piston-valve is shown in Figs. 46 and 47, which is the style adopted in the Forbes high-speed engine. For purposes of computation in designing, the piston-valve may be considered as an ordinary D-valve with its plane surface, and also the rectangular steam-ports, rolled into a cylindrical form. In Figs. 46 and 47, *A* is the circular steam-port through the valve liner, or bushing; *B*



Section through Q.

Fig. 46.



Section through center.

Fig. 47.

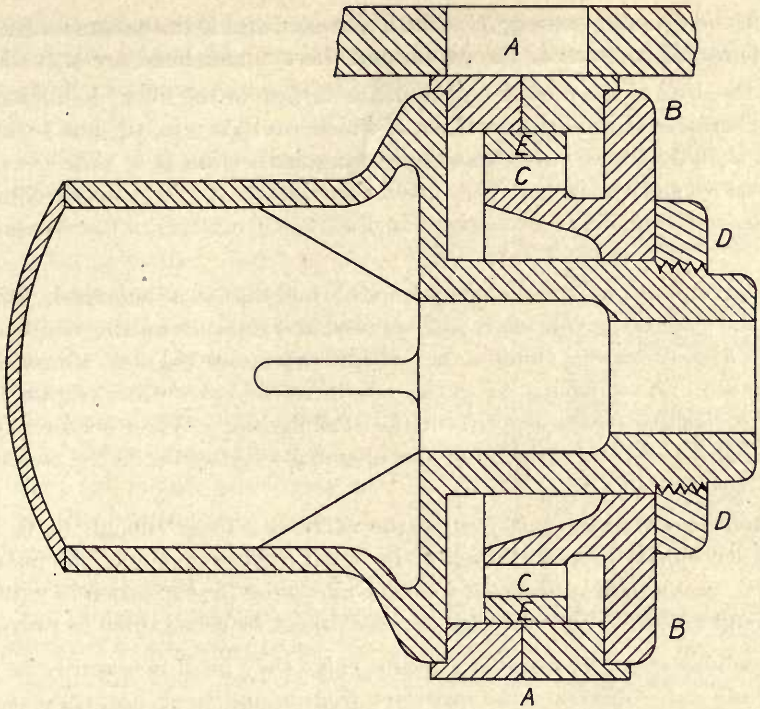


FIG. 48.

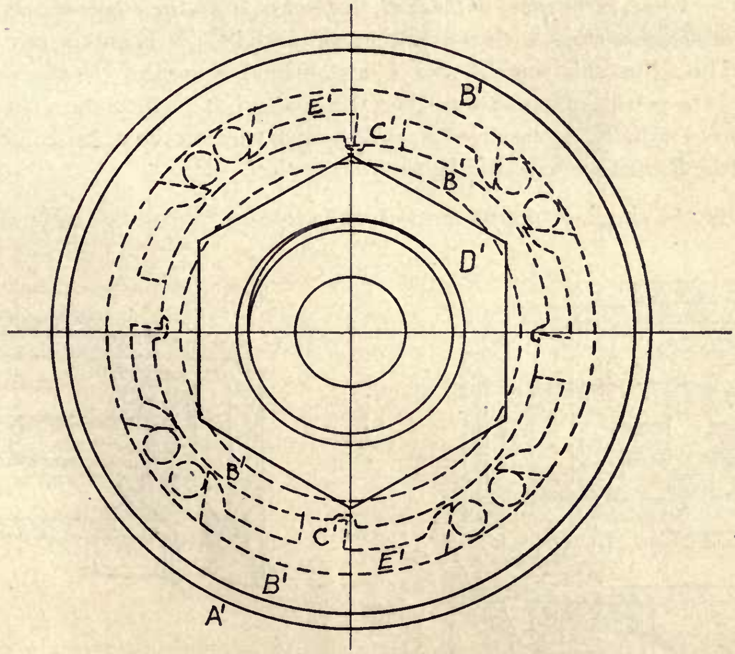


FIG. 49.—End view of Fig. 48.

is the steam-port in the cylinder casting; C is the live steam, and D the exhaust-steam opening in this particular engine. Small portions of the piston and the cylinder head are also shown.

In computing the area of the port A on the *inside* surface of the liner, deductions must be made for the area of the surfaces of the bridges, three of which are shown in the end section view, Fig. 46, at A_2, A_2, A_2. It should be noted also that the rectangular section $(a\ b \times c\ d)$ in the plane R must equal the calculated area of the port. Very often the width of A $(f\ g)$ is made equal to $c\ d$, and the depth of the piston-valve made accordingly. In Fig. 47 $e\ f$ represents the steam-lap, and $h\ g$ the exhaust-lap.

The principal advantages of the piston-valve are: 1st, that it is balanced; 2d, that the valves are at the ends of the cylinder, giving short steam-ports, and thus minimizing condensation; 3d, that when used on the high-pressure cylinders of multiple-expansion engines, steam may be admitted from the inside, or center, and exhausted on the outside of the valve, thus keeping the high pressure and temperature of the live steam away from the stuffing-box. When so used D, Fig. 47, would become the live steam, and C the exhaust-steam openings; $h\ g$ would be the steam-lap, and $e\ f$ the exhaust-lap.

A disadvantage arises from the fact that if the valve is a loose enough fit to slide easily, it is likely to allow the live steam to leak through. In small engines generally the piston-valve is solid, as in Figs. 46 and 47, and a tight, sliding fit, with due allowance for expansion of valve and valve-seat due to different temperatures—especially in starting—must be relied upon to prevent leakage.

When spring packing-rings are used on a piston-valve they must necessarily be thin, and have a small pressure on the valve-seat to avoid excessive friction and wear, and they are liable to break. Adjustable packing-rings must be carefully handled, or they will bind and strip the valve-gear. When spring or adjustable packing-rings are used the steam and exhaust laps are measured from *the edges of the ring, or rings, to the edges of the port,* instead of from the edges of the valve casting. A piston-valve with adjustable rings is shown in Figs. 48 and 49. It is one of the types used on the "Ideal" engine. The adjustable rings A and A' are turned accurately to the bore of the bushing and then split across to permit of expansion when the head, B, B', is turned, so that the cam surfaces between B and C press radially on the shoes E, and these in turn on the rings A. Holes for a spanner wrench are drilled in B, but are omitted in the illustration.

Piston-valves may be single or double-ported, the same as flat valves. Fig. 51 shows the Arm-

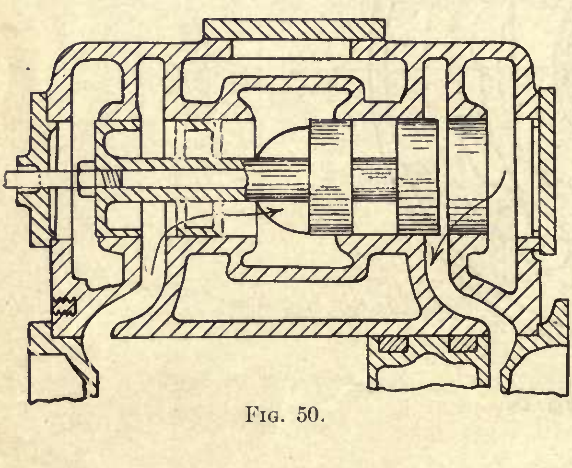


FIG. 50.

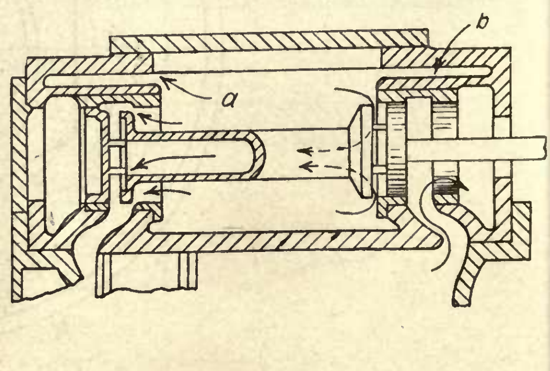


FIG. 51.

ington & Sims type of piston-valve with double admission ports similar to the Allen valve. It will be noted in Fig. 51 how the steam-chest walls at a and b are cored to facilitate uniform heating and cooling of the valve, valve bushing, and adjacent steam-chest wall.

Fig. 52 shows a single piston-valve controlling both the high- and low-pressure pistons of a compound engine as applied to the Vauclain compound locomotive manufactured at the Baldwin Locomotive Works, Philadelphia, Pa. Both pistons move in the same direction at the same time and are secured, through their rods, to the same cross-head. Live steam enters the steam chest at A, A, and passes, at the phase shown in the illustration, through the head-end port to the high-pressure cylinder. The exhaust steam from the high-pressure cylinder is flowing in the direction of the arrows through the hollow center, B, of the piston-valve body to the head end of the low-pressure cylinder. The exhaust from the low-pressure cylinder passes out through the exhaust port, C, to the smoke stack. For the sake of clearness, this illustration is distorted to the extent that the center-line of the steam chest and valve is shown in the plane of the two cylinder center-lines, whereas, to save space in locomotive construction, it is placed back of this plane. It will be noted that the valve is really two piston-valves combined, one formed by the outer rings marked D D, controlling the high pressure and the other formed by the inner rings marked E E, controlling the low-pressure cylinder.

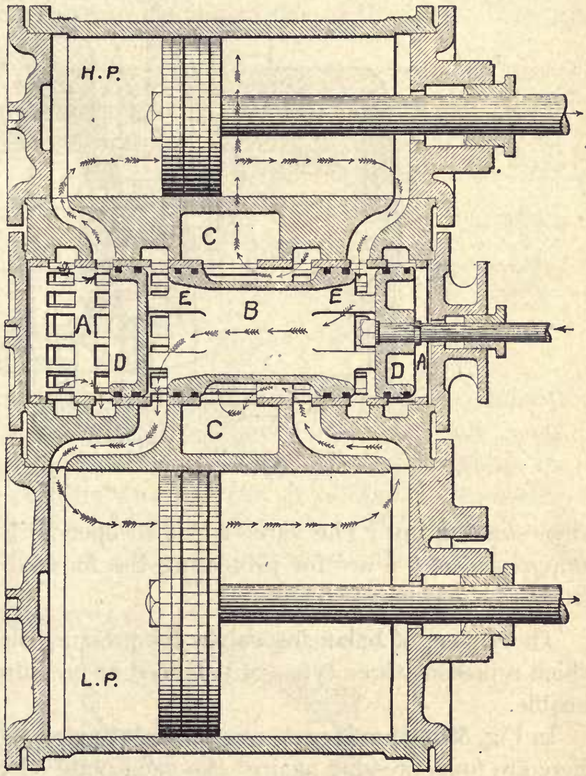


Fig. 52.

Another type of piston-valve, with double-ports and adjustable packing-ring, as used on the Fitchburg engine, is shown in Fig. 53. Four of these valves are used on the engine, the two live-steam valves being operated by a cam-wrist plate and regulator automatically controlling the cut-off, and the two exhaust valves from a separate eccentric. The live-steam valve for the head-end port is shown in the illustration. The action of the valve and method of adjustment will be apparent on inspecting the drawing, it being specially pointed out that the cone, b, has a small clearance at the right-hand end and may be adjusted and locked in any position by releasing the tension bolts, d d, and tightening the compression bolts, e e, against the lugs c of the cone. The expansible ring, a, a, a, a, with its parts joined by radial ribs (not shown), is split on a line parallel with the piston rod and junction strips are set crosswise to keep the steam from passing through.

Pressure-Plate Valves.

The pressure-plate type of balanced valve may be shown by the valve used on the "Straight Line" engine designed by Prof. Sweet, in Fig. 54. A A is the pressure-plate which receives the

pressure of the steam. It is prevented from pressing on the valve by "distance-blocks" (not shown), slightly thicker than the valve which slides between the seat and the plate, and is thus relieved of all pressure. The distance blocks are exposed to live steam the same as the valve in order to prevent binding of the valve between the pressure-plate and seat by expansion, especially

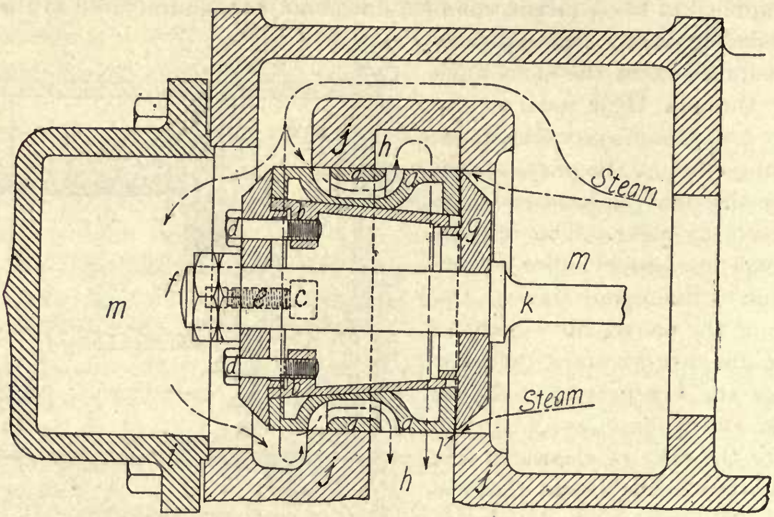


FIG. 53.

when starting up. The valve is shown open to lead on the left end. It is double-ported. The projections at *b b* are for protecting the finished surface of the pressure-plate from the cutting action of the exhaust steam.

The system of balancing valves by pressure-plates is more completely shown in Figs. 55–57, which represent three types of balanced or pressure-plate valve—the fixed, the adjustable and the flexible.

In Fig. 55, steam is prevented from acting on the top of the valve by the bars *a, b,* etc., of which there are four, pressing against the fixed plate *c* by means of springs in the bottom of the grooves which hold the bars.

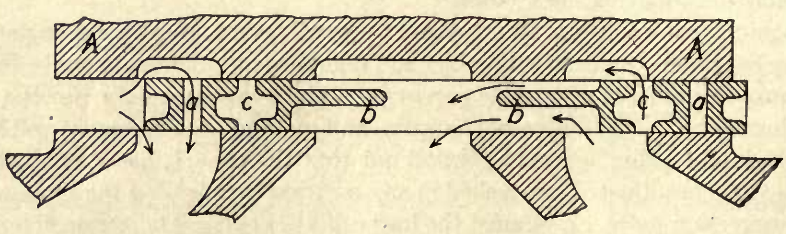


FIG. 54.

In Fig. 56, steam is prevented from acting on the top of the valve by means of the adjustable pressure-plate shown in the background at *b c e f*, which is bounded on the top by the incline plane *b c* and moved or adjusted by the rod *a*. This incidentally shows a type of double-ported valve, similar to the Allen valve, there being a cored passageway (not shown, except as indicated by arrows) from one side of the valve to the other.

In Fig. 57, steam is prevented from acting on the top of the valve by means of the flexible plate, *a*, of steel or other elastic metal so designed as to allow the steam pressure in the steam chest to force the bands *d* and *e* down on the top of the valve with only sufficient pressure to prevent leakage.

In the fixed type of pressure-plate valve, live steam is sometimes relied upon to supply the pressure exerted by the springs as described in connection with Fig. 55, the live steam being admitted through small openings bored in the body of the valve to the under side of the bars. The bars themselves are sometimes enlarged upon and developed to such an extent as to be unrecognizable as bars, they having cut-off and exhaust edges and controlling the steam the same as the original part of the valve. Such types are shown in Figs. 58 and 59, the former being known as the Ball telescopic valve, manufactured by the American Engine Co., Bound Brook, N. J., and the latter as the flat balanced expanding valve manufactured by the Skinner Engine Co., Erie, Pa.

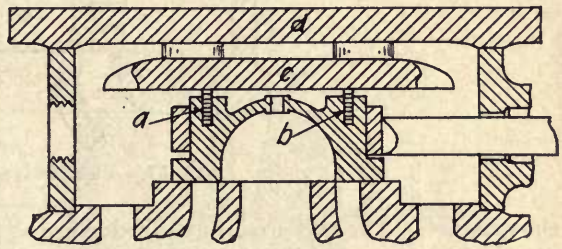


Fig. 55.—Fixed type of pressure-plate valve.

The Ball valve has two rectangular faces and a cylindrical body, the live steam being admitted through the inside, and the exhaust steam passing around the outside edges. The valve seats in the steam chest in this design lie in planes perpendicular to the steam-chest cover plate, these necessitating tortuous steam-ports. This very construction, however, gives double steam admission and exhaust ports and makes the valve in reality a "double-ported" valve with its small valve-travel and without the extra passageway in the valve. The flat balanced valve shown in Fig.

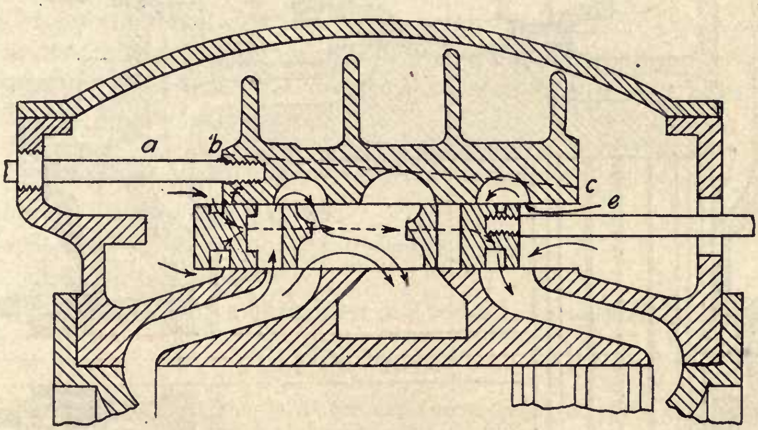


Fig. 56.—Adjustable type of pressure-plate valve.

59 is entirely rectangular and made in two parts with alternate rectangular bars and grooves cut parallel to the port and nearly across the entire width of the valve. The depth of the grooves is a little greater than the height of the bars so that when fitted together there is room for the live steam to flow in. The pressure thus exerted at the twelve spaces similar to the three shown in the upper left half of the Figure at *d*, forces one-half of the valve against the valve-seat and the other half against the face of the steam-chest cover. In the position shown, the first space, counting from

either side, exercises the balancing pressure. As soon as the valve opens to live steam, spaces two and three receive full balancing pressure. When the valve closes for compression the fourth and fifth spaces are subjected to compression pressure. The sixth space always has the same pressure as

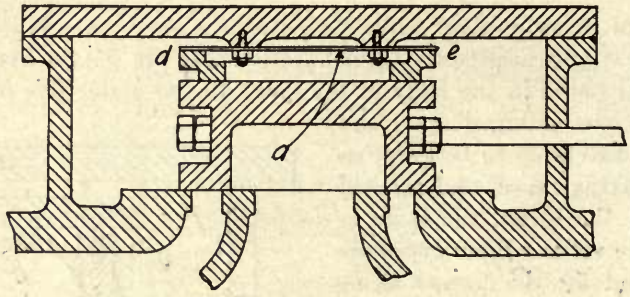


FIG. 57.—Flexible type of pressure-plate valve.

the exhaust. The end areas of the valve are so proportioned that the live steam pressure keeps the two parts of the valve steam tight at e. The valve is double-ported, both for live and exhaust

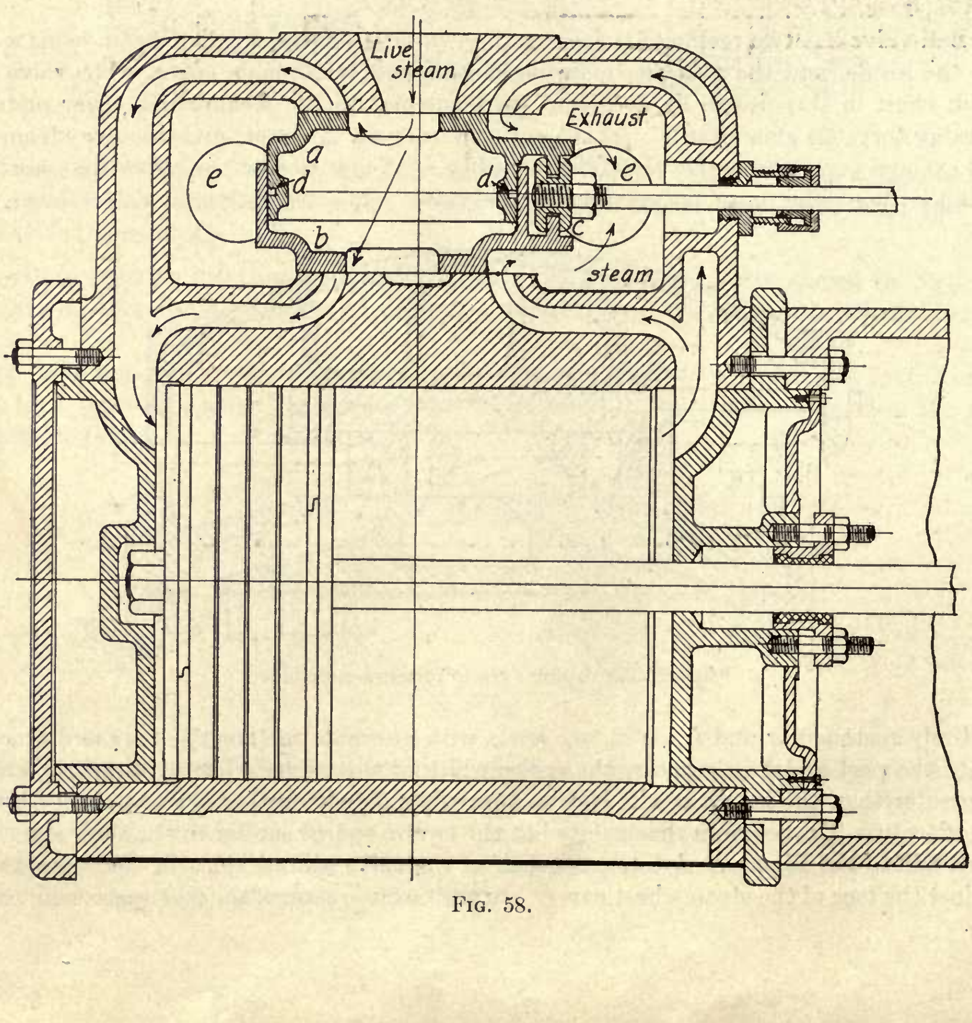


FIG. 58.

steam. The spring at the center of the valve is designed to hold the two parts in correct initial position before steam pressure is applied. The part of the valve marked with the letters *g* and *f*, is an L-shaped hook in which the end of the valve stem engages.

Double-Ported Valves or Their Equivalent.

Prominent examples of the second class of valves are the Allen valve and the Double-Ported valve. The Allen valve, as used in American locomotive practice, is balanced as shown in dotted lines in Fig. 32 of these notes. The double-ported valve is usually balanced. Numerous types

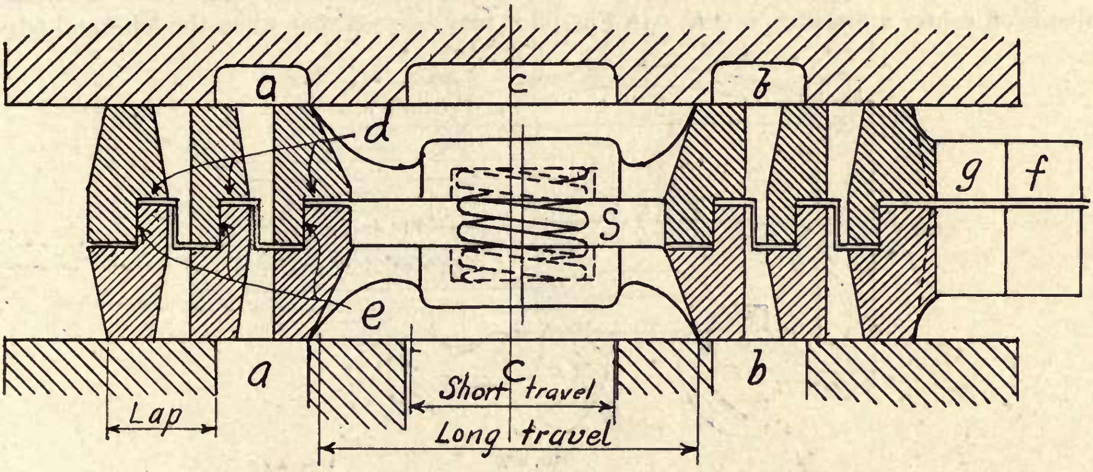


FIG. 59.

of valves have provision for admitting steam to the main port through more than one aperture in the valve, as shown in illustrations already given in the piston and pressure-plate types.

From what has preceded, it appears that many valves of the third class have the combined features of those of the first and second classes.

Valves Which Operate by Two or More Independent Parts.

In valves of the fourth class we will take up:

First, valves divided into two parts known as "double valves."

Second, valves divided into four parts.

Third, valves combining the features of the first and second classes just mentioned.

Two-Part Valves.

In the first class, or in the double valves, there are three distinct types, viz: the Gonzenbach, Polonceau, and Meyer, shown in Figs. 60, 62, and 63, respectively. In each of these three designs the lower valve, or one nearest the cylinder, is called the "main" or "distribution valve." The upper valve is called the "auxiliary" or "cut-off valve." The passageways through the seat in Fig. 60, and through the main valves in Figs. 62 and 63, are called the "auxiliary ports." The two parts of the complete valve are operated by independent gears.

The advantage of valves of this type is that the cut-off may be varied independently of the other events of the stroke, for the reason that the *auxiliary valve affects only the cut-off*, while the main valve governs the admission, release and compression.

Gonzenbach Valve. The Gonzenbach valve has two separate steam chests, A and B, Fig. 60. The partition C has a rectangular, port-shaped opening D. The Zeuner diagram for this valve may be constructed as follows: In Fig. 61 let $E\ F'\ O\ F$ and $G\ O\ G'$ be the Zeuner circles, and $O\ F$ and $O\ G$ the outside laps of the main valve. The exhaust construction for this type of valve is treated entirely in the same way as in the diagram for the plain D-valve, and is therefore omitted here. The angle of advance for the *main* valve may be assumed to be δ.

Let $O\ H$ be the half-travel, ω the angle of advance, which is negative (subtracted from 90°), for the *auxiliary* part in this type of valve, and $H\ K\ O\ L$ the Zeuner circle for the auxiliary valve.

Then for the crank position $O\ I$ the main valve is off center a distance $= O\ J$, and the auxiliary valve is off center a distance $= O\ K$. In Fig. 60 it may be seen that when the left-hand edge of

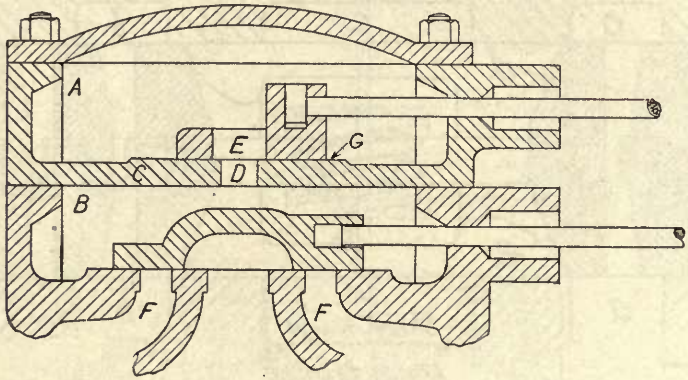


Fig. 60.

the opening E reaches the right-hand edge of the opening D, the main steam chest B is closed, and live steam is cut off from the cylinder even if the main port F is still open. When the opening of the port D becomes zero, the auxiliary valve is off center a distance $= \frac{1}{2} E + \frac{1}{2} D$; this distance is usually designated by S, and is represented in Fig. 61 by the radius $O\ L$ of the arc $L\ M$. Then the auxiliary valve covers the passage-way D while the crank is going from $O\ N$ to $O\ P$, and opens it at $O\ P$ just before the main valve opens the main port at the crank position $O\ Q$ on the return stroke.

In this type of valve, then, the opening of the auxiliary port must always occur after the main port closes on one end (closes at $O\ F\ R$), and before it opens on the other (opens at $O\ G\ Q$). Therefore $O\ P$ must always come between the crank positions $O\ R$ and $O\ Q$. If $\frac{1}{2} E + \frac{1}{2} D$ is made equal to $O\ H$, $O\ P$ will fall on $O\ R$ and the cut-off and main valves will both close at the same time. In this case the half valve-travel of the auxiliary is just equal to $\frac{1}{2} E + \frac{1}{2} D$ and the auxiliary port will be closed for an instant only. If $\frac{1}{2} E + \frac{1}{2} D = O\ T$, $O\ P$ will fall on $O\ Q$, and auxiliary cut-off will take place at $O\ I$, and the auxiliary port will be closed while the crank is turning from $O\ I$ to $O\ Q$. The cut-off therefore is limited between the crank positions $O\ I$ and $O\ R$, and S may have any value between $O\ K$ and $O\ H$.

Should a valve-gear, constructed so that $\frac{1}{2} E + \frac{1}{2} D = O\ V$, have its angle of advance reduced from ω to zero, auxiliary cut-off would take place earlier at $O\ W_1$ and auxiliary port-opening would occur at $O\ V_1$ before the main port had closed at $O\ R$. Therefore steam would be admitted twice on one stroke, illustrating the inadvisability of altering angle of advance without previously determining, by means of a valve diagram, what the effect would be.

In laying out a valve of this kind the area of the main port F, Fig. 60, should be calculated the

same as in the plain D-valve; the area of the auxiliary port D (or ports, as there are sometimes two or more) should be made slightly larger than the area of the steam-port opening and the area of the opening at E in the cut-off valve, or block, should be slightly greater than at D depending

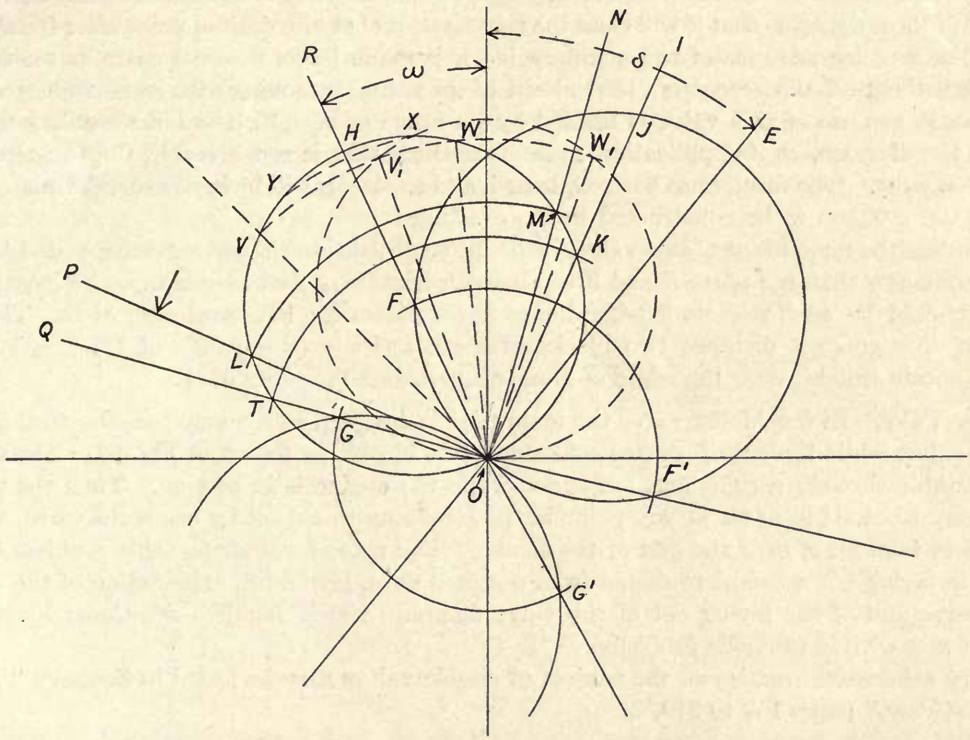


Fig. 61.

on the desired range of cut-off, etc., as found from the Zeuner diagram. The point G of the cut-off valve may be located a distance from the right-hand edge of D = ½ travel of valve + ¼ inch, so that G will never overtravel the port and allow steam to enter at the wrong time.

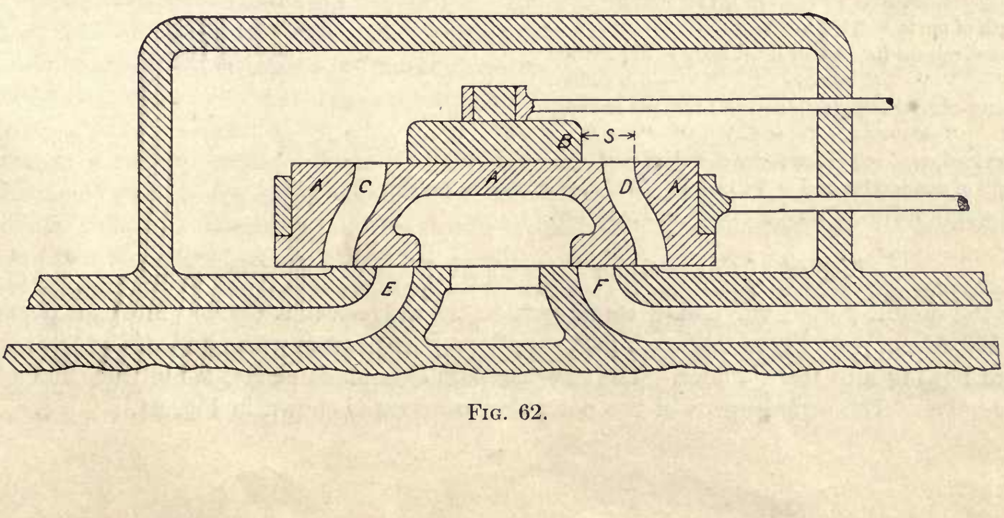


Fig. 62.

In the Polonceau and Meyer valves the auxiliary parts slide on the top of the main block, and both are inclosed in the same steam-chest.

Polonceau Valve. This valve is made in two parts, *A* and *B*, as shown in Fig. 62. The main part *A* is laid out as an ordinary D-valve. The cut-off block *B* is solid and slides on *A*. The motion of *B* is designed so that it will close the passageway *C* at any desired point after *C* has opened to *E*. The angular advance of the auxiliary block is made large, in some cases as much as 90°. Then for half cut-off the eccentric (180° ahead of the crank) is moving with its maximum velocity. Inasmuch as the use of this valve is limited by its range of cut-off, it is not necessary to follow through the diagram, as its application contains nothing that is not given in the Gonzenbach or Meyer diagrams. The former has been explained, and the latter will be in the further notes accompanying the problem to be constructed in the drafting room.

Regarding the dimensions of this valve, it should be noted that the passageways *C* and *D* should be slightly larger than the ports *E* and *F* to allow for friction of flow of steam. The length of the block *B* should be such that its left-hand edge never passes the left-hand edge of *D*. Therefore, length of $B =$ greatest distance two blocks ever get apart $- (S -$ width of D$) + \frac{1}{4}$ inch. In order to obtain smooth wear the edges of *B* must overtravel the edges of *A*.

Meyer Valve. In the Meyer valve the main part is designed in a manner similar to that of the plain D-valve, while the cut-off device consists of two blocks, as shown in Fig. 63. These blocks are adjustable through a right and left screw while the engine is in motion. Thus the point of cut-off may be made to occur at any point in the stroke up to cut-off by the main valve, which is designed to take place near the end of the stroke. The notes for drafting table problem 4, which consists in laying out a complete design from assigned data, give a full explanation of the working of the valve and of the laying out of the valve diagram, which requires additional construction work not met with in previous problems.

A very exhaustive treatise on the subject of double valves may be found in Zeuner's "Treatise on Valve-Gears," pages 159 to 219.

Drafting Table Problem, No. 3.—Double-Ported Valve.

To design a double-ported valve for the low pressure cylinder of the series of U. S. Battleships Nos. 13 to 17 ("Virginia," "Nebraska," "Georgia," "New Jersey," and "Rhode Island").

Bore = 66 inches. Stroke = 48 inches. Revolutions = 120.
Cut-off for top, or head end, = 0.784 of stroke.
" " " bottom = 0.715 " "
Length of ports = $62\frac{1}{2}$ inches.
Exhaust release for top, or head end, = $3\frac{63}{64}$ inches.
" " " bottom = $5\frac{9}{16}$ inches.
Velocity of entering steam = 175.1 feet per second.
" " exhaust " = 125 " " "
Steam lead, top, = $\frac{21}{32}$ inch for each half of port.
Length of connecting-rod = 96 inches. Width of bridge = 2 inches.
Diameter of valve-rod through valve = $2\frac{7}{16}$ inches.

Method of Computation When More Than One Port is Used.

For the double-ported valve each steam-port in the valve-seat is divided into two parts (*m n* and *s t*, Fig. 64, for port T), so that each part supplies a port passage with one-half the total amount of steam flowing into the cylinder. The exhaust port *Q* is made single, being the same as for a plain D-valve. The arrangement of the ports and passages is shown in Fig. 64.

As in the Allen valve, only one-half the steam-port opening need be taken into account in constructing the valve diagram, since the inner port, which gives the other half of the full port-opening, is uncovered at the same time that the outside port is opened.

Calculate the area of the steam-port opening for one end of the cylinder, considering the velocity of the inflowing steam as given in the data for the problem. (In triple-expansion marine work it is common to assume the steam velocity for low pressure cylinder from 6000 to 12,000 feet per minute). Take one-half of this area as the required area to be opened at each port. Divide this by the net length of the ports to obtain the amount which the valve is to uncover the ports for inlet steam. With this port-opening, the proper lead, and the cut-off, construct the Zeuner diagram. The result will be a valve having half the travel of that of the plain D-valve with the same effective opening, lead, and point of cut-off.

In this respect the Allen valve has the same advantage as the double-ported valve, but the Allen valve can only be used with a direct connected eccentric when the points of cut-off are earlier

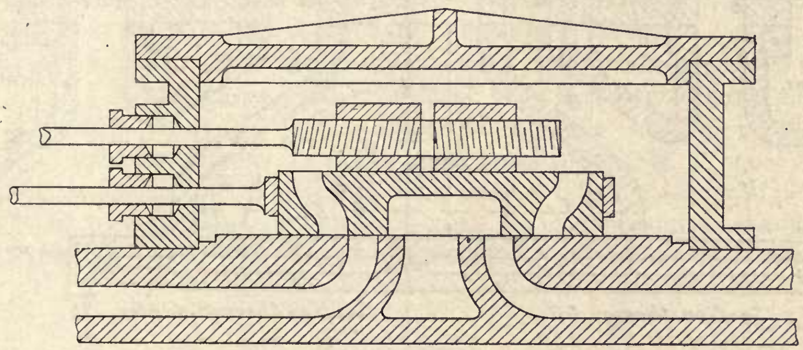


Fig. 63.

than ⅝ stroke; the double-ported may be designed for a broader range of cut-off, but it has, however, a greater area on which the unbalanced steam pressure can act. This disadvantage may be overcome by balancing the valve as shown in Figs. 64 and 65 by packing-rings (as, for example, at E), which are kept firmly against the steam-chest cover H, by means of springs, thus excluding steam pressure from the space G. After the Zeuner diagram is completed for both ends, the various dimensions for the valve are found by the following rules, most of which may be verified by tracing the valve-seat J, Fig. 64, on the edge of a piece of paper and moving the paper the amount of the valve-travel:

The minimum width of bridge ($k\ l$ and $i\ j$) must be such that, for example, the point g of the valve will not under any circumstances come closer than ¼″ to j of the valve-seat. It may be found by the following formula:

Minimum steam-lap + port width + minimum bridge width = half valve-travel + ¼″.

If the result should be a negative quantity, or less than the thickness of the cylinder wall (in this problem, 2 inches) discard it, and make the bridge width equal to the cylinder wall thickness to help insure a sound casting.

Find the width of the exhaust port $j\ k$ which must be such that when the valve is in its extreme position there shall be an opening at least equal to the total steam-port width for one side of the cylinder. This may be found by the following formula:

Width of exhaust port = ½ travel of the valve + maximum exhaust lap + total width of steam-ports for one end − width of bridge.

$f\ g$ = steam-port opening head end; and $o\ p$ = steam-port opening crank end.

The thickness $p\ q$ or $e\ f$, Fig. 64, depends entirely on practical considerations. It is a part of the partition wall, and must be thick enough to give a good casting, and to allow facing. In the present design make it $1\frac{1}{2}$ inches.

The length of that part of the valve-seat ($n\ s$ and $d\ h$, Fig. 64) between the two inlet ports on

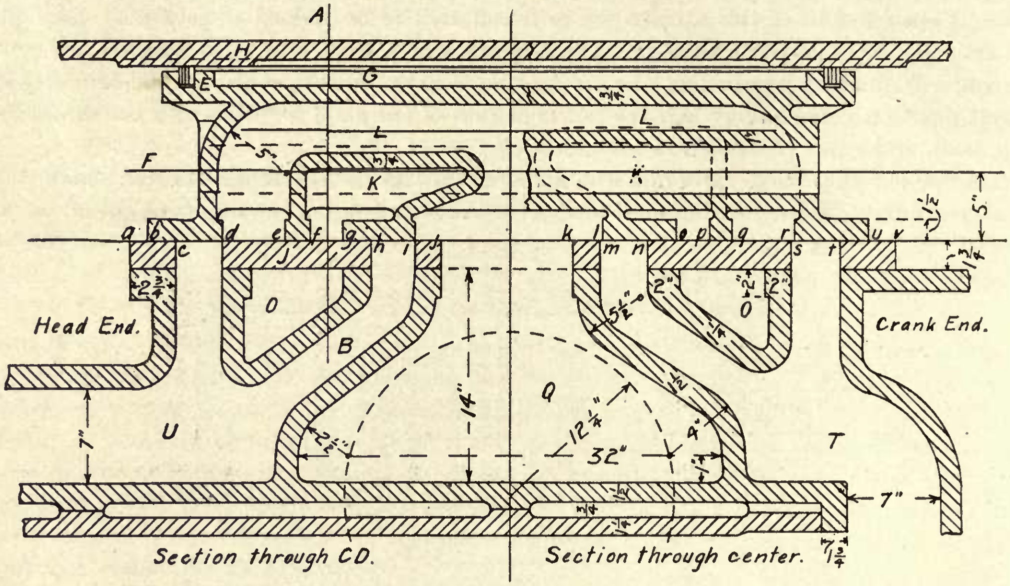


Fig. 64.

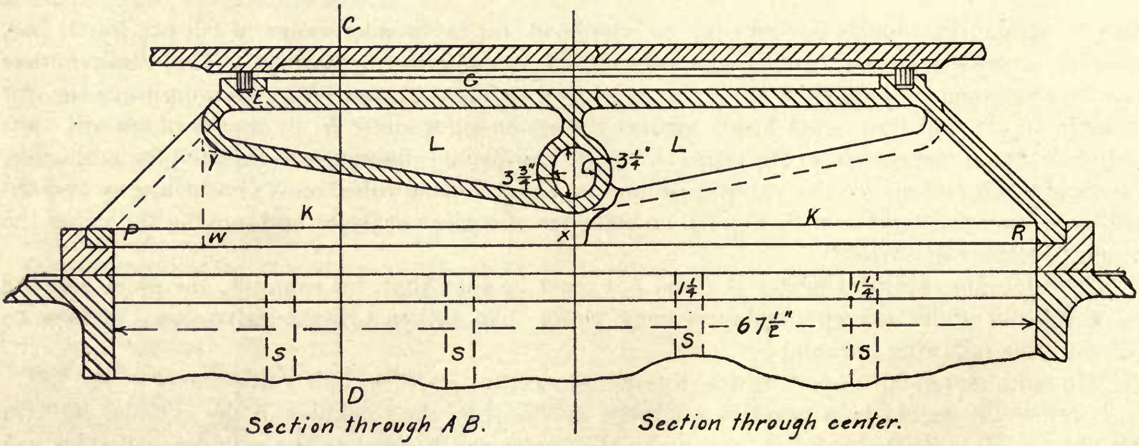


Fig. 65.

each end must be such that the point q does not overtravel s. The proper length is determined by the following formula:

Steam-lap + opening of port + $p\ q$ + $\frac{1}{2}$ travel of valve.

The width of the exhaust passage $d\ e$ and $q\ r$ through the valve, Fig. 64, will, according to the previous paragraph, be equal to $\frac{1}{2}$ travel minus exhaust-lap (according to end for which compu-

tation is being made). Should this give values to $d\ e$ or $q\ r$ less than $c\ d$, it will be necessary to arbitrarily lengthen $n\ s$ and $d\ h$. This can only happen when port width plus exhaust-lap is greater than half valve-travel and will rarely, if ever, occur.

The ports $s\ t$, $m\ n$, $h\ i$, and $c\ d$ are made equal.

The maximum distance for $a\ c$ or $t\ v$ should equal such an amount that the points b or u will overtravel the edge, but not so small that the points d or r will overtravel.

Computation for Steam Passageway in the Valve Itself.

The steam supplied to the inner steam-ports $f\ g$ and $o\ p$ of the valve is conducted through conical pipes from the sides of the valve which are shown at $K\ K$ in Figs. 64 and 65. The usual form of a cross-section of these pipes is shown in Fig. 64. The area of a cross-section of the pipe at $y\ w$ equals the area of the steam-port opening from w to x, less the area of two supporting ribs $S\ S$, w and y being located by trial and error to satisfy this condition. With the point y determined, the slanting lines of the top of the pipe might be drawn directly to x; as the left-hand pipe is not required to feed the right half of the port. It is often drawn from y tangent to the valve-stem casing, as shown in Fig. 65. Make the slope of the side of the valve about 45°.

The right half of Fig. 65 shows a section through the valve at the center, and the left half a section at $A\ B$ through one of the "conical steam passageways" or "pipes," as they are called.

It now remains to make the sum of the two upper areas $L\ L$ in Fig. 65 equal to the area of one of the steam-ports at one end of the cylinder. This is equal to $c\ d \times$ length of port. This is most easily accomplished by considering the areas as made up of approximate triangles.

Make out a table as follows, and enter the results of the calculations therein:

	Top, or head end.	Bottom, or crank end.
Eccentricity..		
Travel of valve..		
Width of port..		
Steam-lap..		
Exhaust-lap..		
Angular advance...		
Steam lead...		
Cut-off, inches..		
Cut-off, per cent. of stroke................................		
Exhaust release in inches...................................		
Exhaust release, per cent. of stroke.......................		
Compression, inches...		
Steam opening..		
Exhaust opening*..		
Velocity of steam, feet per second.........................		
Velocity of exhaust steam, feet per second................		

Area of Exhaust Passageway in Cylinder.

The drawing is to be completed as shown in Figs. 64 and 65. Place the Zeuner diagram and table of results on one sheet, and the valve drawings on another. The principal dimensions for

* Enter the words "full port" unless the exhaust opening is less than the width of the port.

such parts as are not calculated will be found on the sketch. The openings *o o* are merely cored out to save weight, and have nothing to do with the working of the valve. In many cases this part of the valve-seat is cast solid. The area of the exhaust port *Q*, in this case, is made less than the area of the combined steam-ports of one end, allowance being made for the fact that the section is customarily shown in a central plane, and therefore only about half of the exhaust steam has to pass through the section. In this case the area *Q* is about 0.7 of the area of the ports. Immediately beyond the section *Q* the exhaust passageway enlarges, due to the curvature of the cylinder wall, and is ample to conduct the steam to the exhaust pipe, represented by the dotted circle with a 12¼-inch radius.

In Fig. 65 the length of the port is shown as 67½ inches instead of 62½ inches as given in data. This increase is made necessary by the four 1¼-inch supporting ribs shown at *S*.

Drafting Table Problem, No. 4.—Meyer Valve.

In the Meyer valve the upper or auxiliary part is made in two pieces, as indicated in Fig. 68 by *R* and *P*, and the cut-off may be varied at will while the engine is in motion by moving the two pieces nearer together, or farther apart, by means of the hand-wheel *O*, shown in Fig. 68. The top view of the main valve, which in this case is divided into two portions, is shown in Fig. 67.

The present problem consists in designing a Meyer valve for a steam air compressor of the following dimensions:

Stroke	30 inches
Bore	16.5 inches
Revolutions per minute	60
Maximum cut-off of main valve (head end)	
Lead of main valve (each end)	0
Inside lap (each end)	0
Velocity of live steam	6000 feet per minute
" " exhaust steam	4000 " " "
Length of port	13.5 inches
Connecting-rod	5 crank lengths
Range of cut-off for auxiliary valve	

In the solution of the problem, find first the maximum port-opening required, and then, by means of an ordinary Zeuner diagram, find the outside lap of a plain D-valve that will give the desired maximum cut-off.

As shown in Fig. 68, the live steam must pass through the opening *t n b c;* hence *b c* will be equal to the port-opening, and inasmuch as the two parts of the valve are not shown on center the steam-lap will not show directly but will be equal to *c v - B F*. In the drawing, the valves are shown in a proper working position for cut-off, approximately at half stroke (at $C\ V_1$, Fig. 66). The scale of the Zeuner diagram, Fig. 66, is between four and five times the scale of Fig. 68. The exhaust cavity of the valve is sometimes divided into two parts as shown.

To Find the Auxiliary Valve Circles C K and C L.

Place the cut-off valve eccentric about 45° in advance of the main-valve eccentric.

Make the travel of the auxiliary valve in this problem 3 inches. (The travel of the auxiliary valve in Fig. 66 is represented by *L K*.)

To Find the Relative Valve Circle Showing How Far the Two Valves are Apart at any Instant.

Draw the line *O K* joining the extremities of the diameters of the Zeuner circles for the main and auxiliary valves. Through the center of the diagram, *C*, draw a line *C G* parallel and equal to *O K*. Upon this line as a diameter describe a circle *C B H I* passing through the center. This

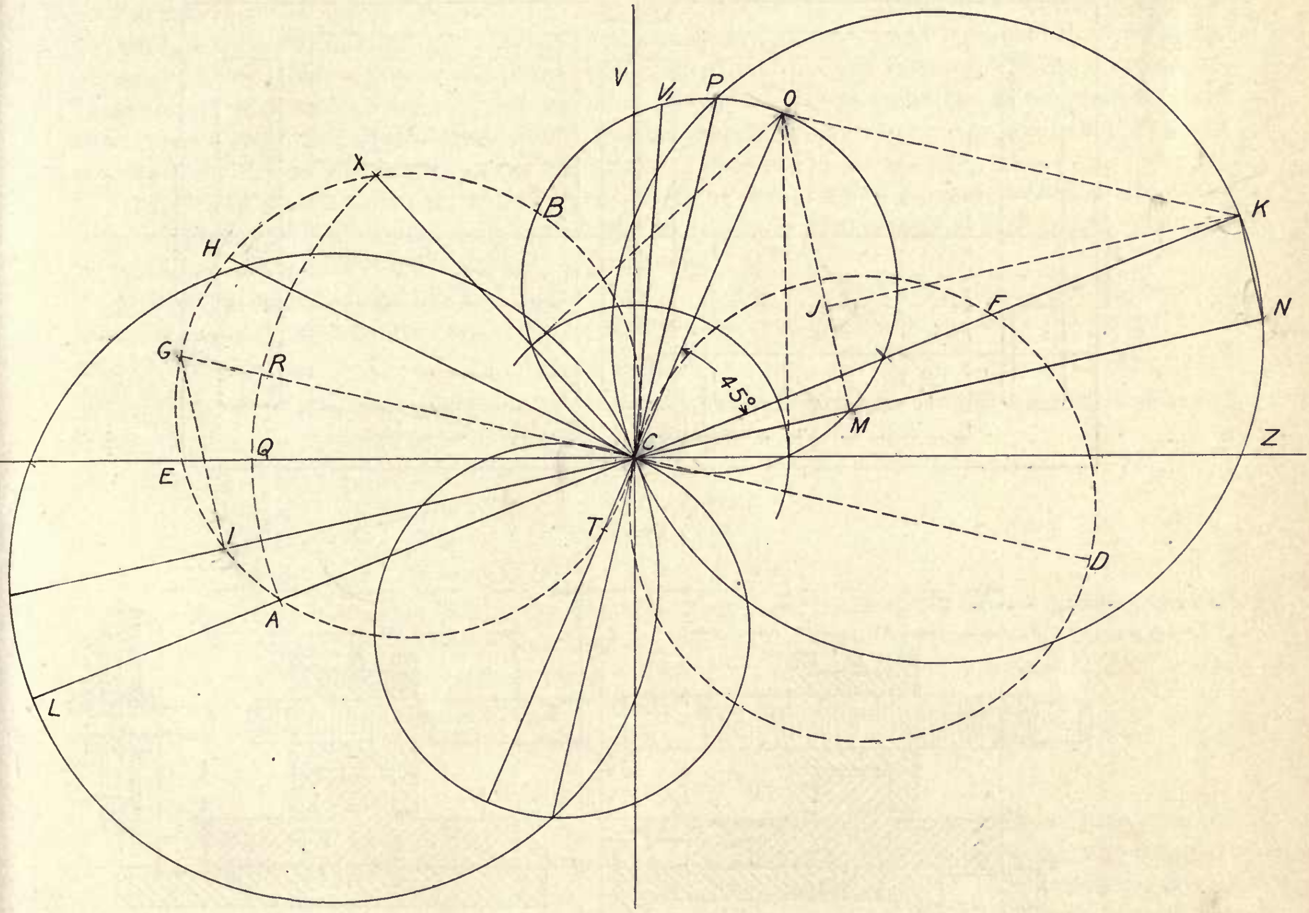


Fig. 66.

is called the "Relative Valve Circle," and shows for any position of the crank the amount the two valves are apart, as the following example will show:

Suppose the crank at *C N*. Then the main valve is off center the distance *C M*, and still going farther away; the auxiliary or cut-off valve is off center the distance *C N*, and also going farther away.

Hence the centers of the two valves must be the distance *M N* from each other. But if the relative valve circle *C B H I* shows the relative positions of the two valves for any crank position, then the distance *C I* should equal *M N*, which it does. This may be shown by the similar triangles *O K J* and *G C I*, the line *J K* being drawn parallel and equal to *M N*. A similar relative valve circle, *C F D*, may be used for the analysis of the stroke on the opposite end of the cylinder.

Explanation of the Value of S Which Determines the Point of Cut-Off.

Let S = the distance from the opposite edge of the block (a' k', Fig. 68) to the cutting-off edge of the main valve (t b) when the two valves are central with respect to each other, or $S \doteq$

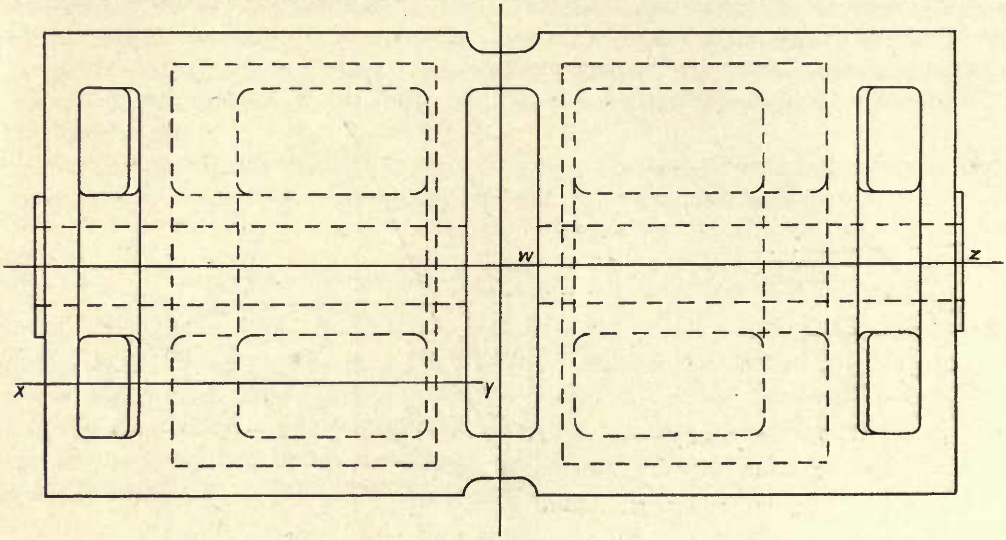


Fig. 67.

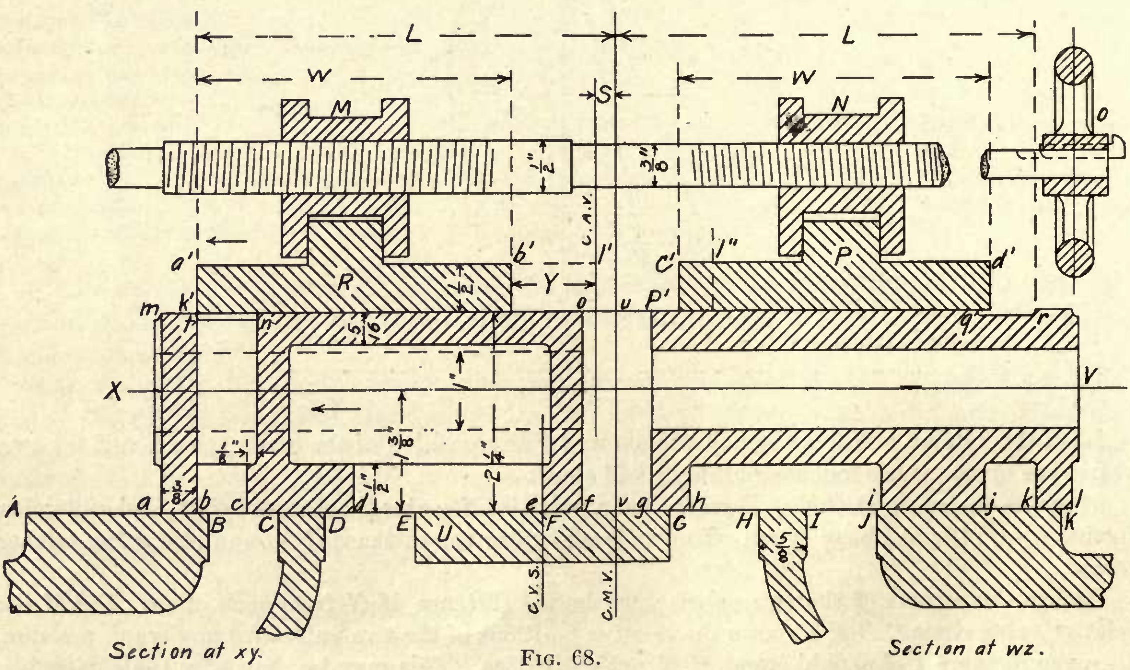


Fig. 68.

the distance between the main and auxiliary valve center-lines when the auxiliary block is in its cutting-off position as shown in Fig. 68.

Then S = the distance from t to the main valve center-line minus the distance from a' to the auxiliary valve center-line = $t\,u - a'\,l'$. If the latest auxiliary cut-off be assumed when the crank is in the position $C\,H$ (Fig. 66), the blocks R and P (Fig. 68) may be designed so that they are zero distance apart (the points b' and c' will then be at l'), at which time S is a maximum and equals $C\,H$ in Fig. 66. All earlier cut-offs may be obtained by separating the blocks and thus reducing the value of S.

For cut-off at $C\,P$ (perpendicular to $C\,G$, Fig. 66), S would be zero because $C\,P$ has no intercept in the relative valve circle, both valves being the same distance off center and consequently having zero distance between their center-lines. For cut-off earlier than this, the value of S would be negative, being measured on the extension of the crank line, as $C\,T$ and $C\,I$ at crank cut-off positions $C\,O$ and $C\,N$, respectively; and these values would appear as auxiliary laps, or the amount that k' would overlap t, Fig. 68, when the two valves are relatively centered.

The earliest possible auxiliary cut-off would be at the main valve admission, which in this problem (there being no lead) would occur at $C\,Z$, Fig. 66. Then S would equal $-\,C\,E$, and the distance between the blocks (call it $2\,Y$) would be a maximum.

If W = the width of the blocks R and P, and Y = the distance the inside edge of the block is from the auxiliary valve center-line (= $b'\,l'$ for position shown in Fig. 68), we have for the general case,

$$L = S + W + Y \qquad (1)$$

While L and W remain constant S and Y vary for different cut-offs, as the following cases, Fig. 66, will show, but the algebraic sum of Y and S is a constant:

For cut-off at $C\,H$, Fig. 66, $S = C\,H$, and $Y = O$.
" " " $C\,X$, $S = C\,X$, " $Y = C\,H - C\,X$.
" " " $C\,P$, $S = O$, " $Y = C\,H$.
" " " $C\,Z$, $S = -\,C\,E$, " $Y = C\,H + C\,E$.

Therefore the latter value of Y is the greatest it can have between the above limits of cut-off. By substituting any of the corresponding values of Y and of S above in equation (1), the value of L can be obtained. Taking the corresponding values of Y and S for cut-off positions $C\,Z$:

$$\begin{aligned} L &= -\,C\,E + W + C\,H + C\,E \\ &= W + C\,H \qquad (2) \end{aligned}$$

Width W of Cut-Off Blocks.

The relative valve motion should never be so great that the inside edge of the block uncovers the inlet passage, $t\,n\,b\,c$, Fig. 68. To obtain a width to insure this at all times, it is necessary to consider:

1. The very earliest cut-off when the crank is in the position $C\,Z$. Y then has its maximum value, and the edge $a'\,k'$, Fig. 68, is directly over the edge $t\,b$, and the center of the auxiliary valve the distance $C\,E$ (Fig. 66) to the right of the main valve center, or in position shown by dotted line l''. The outside edge $a'\,k'$ would then have to move the distance $C\,E$ beyond t before the two valves are again centered with respect to each other.

2. After the valves are centered, allowance must be made for the maximum distance the valves move apart, $C\,G$, which distance the edge $a'\,k'$ may move still farther beyond t.

3. The edge $a'\,k'$ has now moved the distance $C\,E + C\,G$ beyond t, and the width of the block must be sufficient to equal this and also cover the passageway $t\,n$.

4. In addition, the block in its extreme position should still have a small amount overlapping the edge n. One-quarter of an inch may be allowed for this.

To sum up, $W = C\,E + C\,G + t\,n + \frac{1}{4}$ inch.

By substituting this value of W in equation (2),

$$L = C\,E + C\,G + t\,n + \tfrac{1}{4}\text{ inch} + C\,H \qquad (3)$$

Hence, $2L$, or the distance from t to r (Fig. 68), equals $2(CE + CG + CH + \frac{1}{4}$ inch$)$ + 2 width of steam-inlet passageway. The width $t\,n$ of the steam-inlet passageway may be made ⅛ inch greater than the width $b\,c$ of the steam-port opening.

In drawing the valve for this problem place the blocks so that cut-off occurs at. . . .stroke. The valve drawing in Fig. 68 is laid out to correspond approximately with the Zeuner diagram, Fig. 66, for cut-off when the crank is in the position $C\,V_1$.

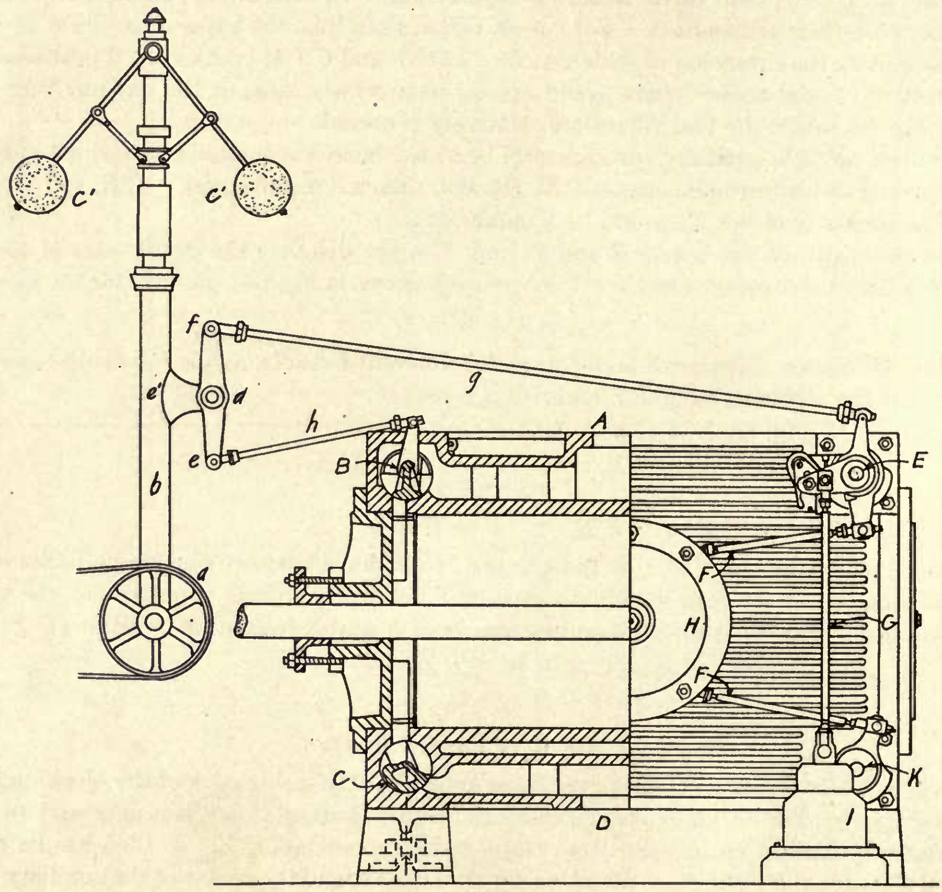


FIG. 69.

The dimensions shown on the valve in the accompanying drawing are to be used in laying out the drawing for this problem. Students may omit the top view.

In designs for this valve the value for the distance $b\,k$ may come out so large that the width of the exhaust-port is excessive; sometimes there is room to divide the exhaust-port and the valves in two parts, as shown in the drawing. Should there not be room in the present problem to permit of this division, omit the part U and run the two exhaust cavities under the main valve into one. Or, as the problem permits, the passageways $t\,n\,b\,c$ and $r\,q\,j\,k$ may curve either towards or away from the center.

In the finished drawing place sufficient dimensions for a working design. Place the points A and K of the valve-seat so that the edges a and l of the valve will overtravel ¼ inch.

In valves of the 2d class, which are operated by four independent parts, the most important type is the Corliss valve, shown in Fig. 69.

Corliss Valve-Gear.

The original type of valve-gear, known as the "Corliss," was patented by Mr. G. H. Corliss of Providence, R. I., in 1849. This type of gear was subsequently taken up by numerous manufacturers, who substituted various alterations in details of the design, until now such names as Harris-Corliss, Allis-Corliss, etc., are common.

Fig. 69 shows a Corliss engine, one-half in section and one-half in full front view. The names of

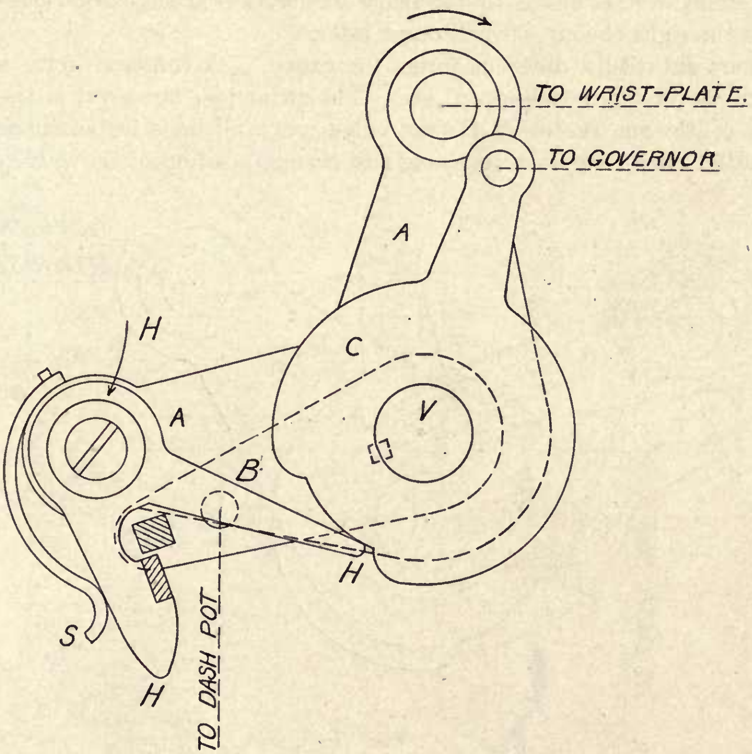


Fig. 70.

the parts operating on the valve-stem at E are shown in Fig. 70, on an enlarged scale. The other parts are

A, steam-pipe opening.	B and E, steam-valves.	C and K, exhaust-valves.
D, exhaust-pipe opening.	F, radius-rods.	I, dashpot.
G, dashpot-rod.	H, wrist-plate.	

The detail of the dashpot will be shown in a later sketch.

The Corliss valve may be considered as a plain D-valve with its outside and inside laps separated into four independent parts, and one placed at each corner of the engine cylinder.

Advantages: Short, direct passages reducing steam clearance; reduced valve motion, each valve being designed to move only a little after port is closed and then remain at rest until time to open again.

Detail and Operation of Releasing Gear.

Fig. 70 is for valve about to open, Fig. 71 for valve about to close. *A* is a bell-crank lever mounted loosely on the valve-stem. *V* is the valve-stem. One arm of *A* carries a pin for rod to wrist-plate; the other a pin on which swings freely the grab-hook *H*. Hook *H* is pressed in by spring *S* so that one arm is always held firmly to knock-off cam *C*, which is also loose on the valve-stem. Cam *C* has an arm to which is attached the governor-rod (see *g*, Fig. 69) to the governor. Drop lever *B* is keyed to the valve-stem and connected to the dashpot by a rod; it also carries a steel block, or die, which engages with the die on the grab-hook arm.

A movement of the cam *C* to the left by the governor-rod will cause the hook to strike the cam-projection earlier and the steel blocks to disengage sooner, thus giving earlier cut-off; when the governor-rod pulls to the right the cut-off will occur later.

Different makers substitute different forms for cam *C*. A common form is to have a plain hub with steel knock-off block bolted on, etc. The principle, however, is the same in all.

Fig. 72 shows center-line sketch of Corliss valve-gear; full lines for eccentric in extreme left-hand position; dotted lines (No. 2) for opening and closing position of the valve, and lines (No. 3)

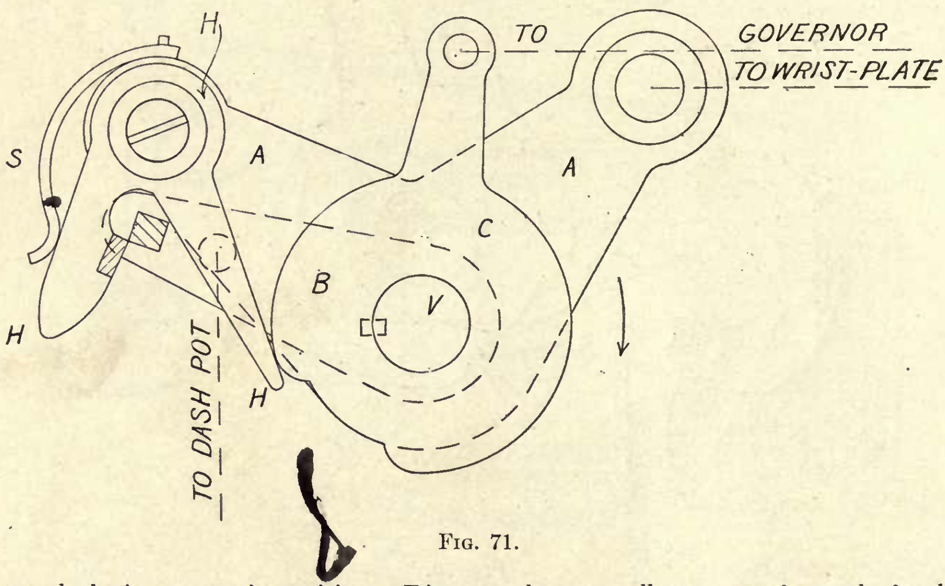


Fig. 71.

for extreme clockwise eccentric position. Diagram shows small amount of travel of valve when port is closed as compared to that when open by the angles marked "closed" and "open." This is an essential feature of the valve; giving as it does quick travel at steam-port opening, and reducing its entire travel to a minimum. The adjustment of the angles for the valve arms and rods is a matter of trial-and-error until desired results are produced; they may be placed above the valve (as shown in Fig. 72), or below, (Fig. 69) according to the circumstances surrounding the design.

Angles *a* are the same, radial line *f* passing through edge of port, and similar line *g* through edge of valve for extreme closed position. Sectional and dotted outlines show extreme positions of valve. Exhaust valve has a positive motion, i.e., has no automatic releasing gear as has the steam valve.

Corliss valves may be single-ported (Fig. 72), or double-ported (Fig. 73).

Limited Range of Cut-Off With Single Eccentric.

The Corliss valve-gear *with single eccentric* will operate the cut-off automatically only up to half stroke as the following argument will show:

When release occurs, the main crank has not quite reached the dead-point; also, when compression occurs, the crank has not reached the other dead-point. When the crank is half-way between the positions corresponding to release and compression, it is still short of the 90° position, and the

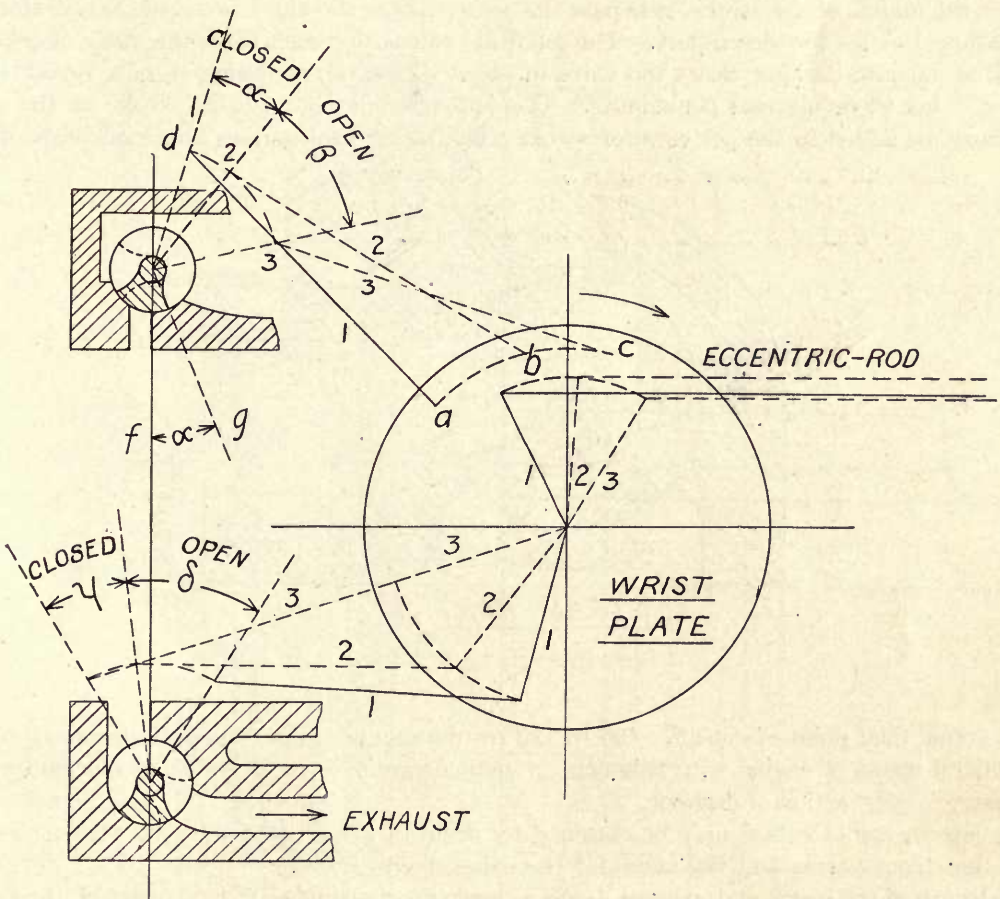


FIG. 72.

piston is short of half-stroke. When the crank is in this "half-way" position, the exhaust-valve, the exhaust-valve radius-rod and the wrist-plate are all at extreme throw, for the exhaust-valve is in exactly the same positions at release and compression, and its motion comes indirectly from the main crank shaft. When the wrist-plate is at extreme throw the grab-hook is in its highest position and if it does not strike the governor cam by the time it reaches this highest position it will not strike it at all.* But it has been seen that the wrist-plate (and therefore the grab-hook) reaches its extreme

*In this event the blocks on the grab-hook and valve-stem arms will remain in engagement during the entire cycle and cut-off will occur at or near the end of the stroke.

throw before half-stroke. Therefore, automatic cut-off by the dashpot cannot occur later than half-stroke. When this is understood it will be seen that the exhaust steam requirements at one end of the cylinder actually control the latest point of cut-off on the other end of the cylinder when a single eccentric is used.

In good indicator cards from fast running, single eccentric, Corliss engines, it sometimes *appears* that cut-off takes place later than half stroke. This is only apparent, however, as the cut-off actually *begins* before half stroke, but the relatively fast moving piston, which is at its maximum at about the middle of the stroke, gets past the center before the valve (even when operated by a good dashpot) closes the steam-port. The following calculation may show this more clearly:

A good vacuum dashpot closes the valve in about $\frac{1}{16}$ second, or during $\frac{1}{10}$ of a revolution for engine running 96 revolutions per minute. This is approximately $\frac{1}{5}$ of the stroke at the center, which must be added to the per cent. of stroke when steam hook strikes knock-off block or cam,

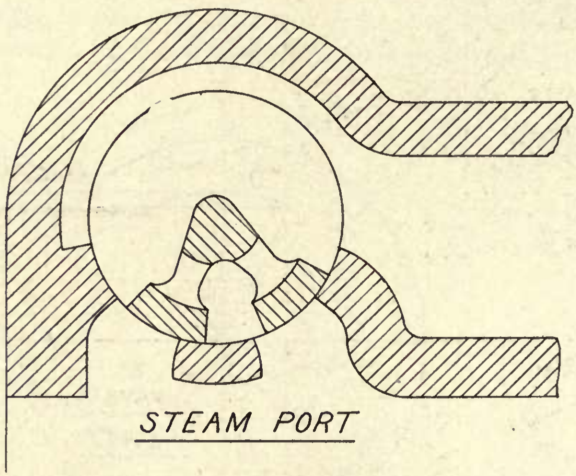


FIG. 73.

to give actual final point of cut-off. 100 to 120 revolutions per minute may be assumed as practical limit of speed of engine with this type of gear, owing to wear of releasing mechanism, and comparatively slow action of dashpot.

A greater range of cut-off may be obtained by using two eccentrics and two wrist-plates, one set for the steam valves and the other for the exhaust valves.

The length of the steam and exhaust ports is made nearly equal to the diameter of the cylinder bore, as a rule. Steam and exhaust valves are usually made of equal diameter, and vary from $\frac{1}{5}$ cylinder bore in small engines to $\frac{1}{6}$ in larger ones. For steam-port, a steam velocity of 8000 feet per minute may be allowed; for exhaust-port, 6000.

Setting Corliss Valve-Gear.

"Adjust length of eccentric-rods to give wrist-plate equal travel on both sides of center mark on bracket. Adjust radius-rods to give proper lap with wrist-plate in central position. Move wrist-plate to end of its travel either way, and adjust length of dashpot-rods to let hooks engage freely on catch-blocks. Put crank on dead-center, and set eccentric ahead of crank enough to give desired lead. Raise governor to highest working position, and adjust length of governor-

rods so that the knock-off cams will just keep hooks off catch-blocks—or some initial motion may be allowed, but not enough to open the port."

Dashpots are of various forms and construction, the principle in all cases being that a vacuum is used to accelerate the fall of the plunger, or bell (*A*, Fig. 74). An air cushion is provided to bring the plunger to rest without shock. Fig. 74 represents a well-known design. In this case *C* is the dashpot-rod leading to the drop lever which, in turn, is keyed to the valve-stem. The

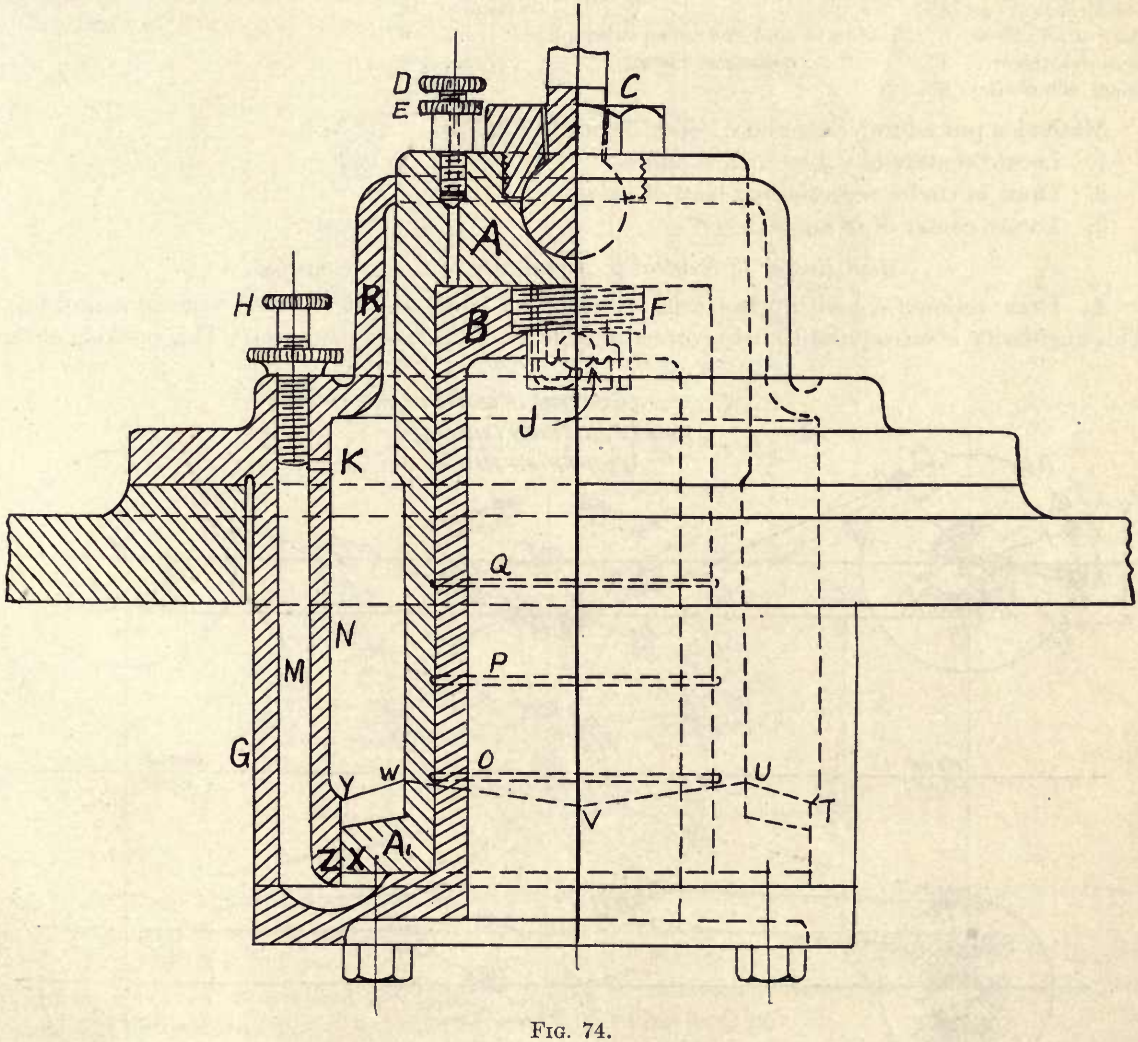


Fig. 74.

ball joint is used to accommodate a slight swing of the rod. The vacuum is created between *A* and *B*. *B* may be termed a "stationary piston." *D* is a regulation screw with locknut *E*. Through *D* is drilled a small hole with side outlets just above the cone seat, as shown. Any air pressure which might accumulate in the vacuum space is expelled through small drilled holes leading to the under side of the ball seat at *F*. The air cushion is formed and acts while the point *x* of the flange A_1 of the plunger is passing from *y* to *z*. The cushioning effect is adjusted by the solid

thumb screw H, which regulates the flow of air from the passage M through the opening K to the space N. G is the body, and R the cover of the dashpot. The circular grooves O P Q, etc., are for lubrication. The sloping edges t u v w y are designed to prevent a too sudden cushioning effect.

Drafting Table Problem, No. 5—Corliss Valve-Gear.

Data for Problem:

Bore of cylinder . . 12″.	Distance between centers of valves (horizontal) 32″.
Stroke . . . . 24″.	" " " " " (vertical) 16″.
Bore of all valves. 3″.	Radius to hook-rod pin on swing-plate . . . . 8″.
Eccentric throw . . . 3″.	" radius-rod pin on " . . . . 6″.
Diam. of hub circle 5″.	

Method of procedure: (Reference letters belong to Fig. 75).

1. Locate centers of valves a, b, c, and d.
2. Draw in circles representing bore of valves.
3. Locate center of swing-plate(e).

Bent Rocker to Neutralize Angularity of Connecting-Rod.

4. Draw rocker f e g with upper arm vertical, and lower one at 15° with vertical center-line. This angularity is introduced to help correct angularity of the connecting-rod. This position of the

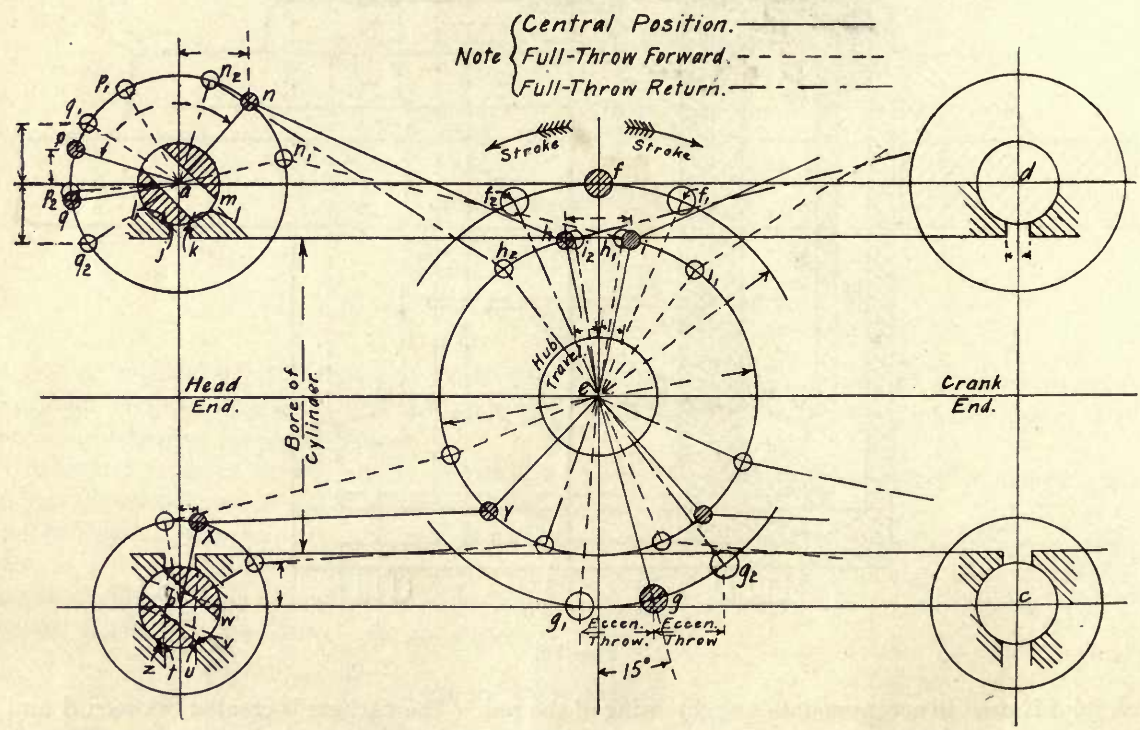


Fig. 75.

rocker is its central position corresponding to zero throw of the eccentric. The rocker f e g in practice (in long-frame engines) is placed at some convenient point between the cylinder and the shaft, the eccentric-rod connecting to the point g, and one end of the hook-rod to f. The other end of the

hook-rod is attached to a point on the swing-plate occupying the same position as f. The points, h, i, y, etc., are also on the swing-plate. The rocker, in designing, is shown attached to the swing-plate shaft for convenience and to save space in the lay-out.

5. Show rocker-pins f and g with a diameter of 1 inch, and draw indefinite arcs on which the extreme travels of f and g will be shown later.

6. Lay off eccentric-throw from g, and locate extreme positions of rocker-pins for full throw forward stroke (g_1 and f_1) and full throw return stroke (g_2 and f_2).

7. Draw all arms, links, etc., in solid lines, and cross-section all circles, for the zero eccentric-

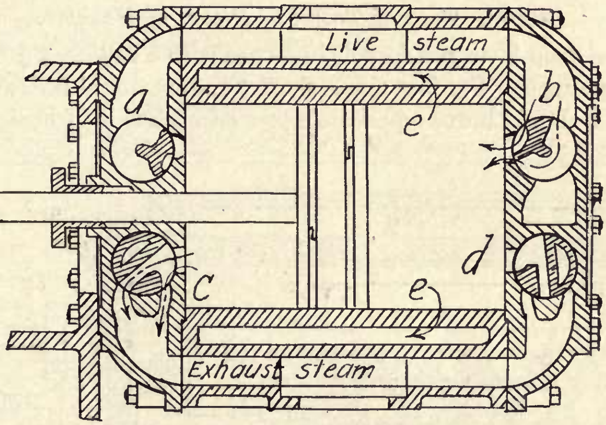


FIG. 76.—Section of Atlas Cylinder.

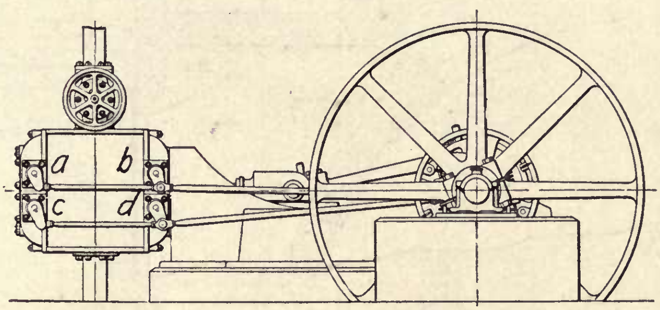


FIG. 76a.—View of Atlas Engine.

throw position. For arms, links, etc., in full throw forward and return stroke positions use characteristic lines shown in sketch, and leave circles open.

8. Locate radius-rod pins h and i on swing-plate 3 inches apart. This is the minimum distance to allow for machining and play between the rod ends. Make pins h and i $\frac{3}{4}$ inch in diameter. Show these pins in extreme forward and return positions.

9. Make steam-port width, $j\,k = \frac{7}{8}$ inch.

10. Make steam-lap, $k\,l = \frac{5}{32}$ inch.

11. Make width of passage through valve, $l\,m = 1\frac{1}{8}$ inches.

Determination of Valve Travel for Cylindrical Valve.

12. Find, by method of trial-and-error, the point n of the rod $n\,h$ and rocker-arm $n\,a$ so proportioned as to turn point l of valve to l_1 (l travels a small distance, say $\frac{1}{16}$ inch, beyond j so as to

produce more uniform wear on valve and valve-seat) while h travels to h_1. Mark the corresponding extreme point of travel of n at n_1; also mark the other extreme point of travel for n at n_2 when h reaches h_2. The lines $a\ n_1$ and $n_1\ h_1$, and also $n_2\ h_2$ and $h_2\ e$ must not be allowed to reach a straight line position. This trial-and-error process is usually accomplished by a stiff paper model on a full scale; but as a student exercise it may be done with two pairs of dividers, or with dividers and compass. The length of the arm, $a\ n$, may be assumed as 4 inches.

13. Locate points on crank end of steam-valve corresponding to the points n, n_1 and n_2. The same arguments and methods apply, but the results are slightly different.

Determination of Travel of Piston of Dashpot.

14. Assume that the drop-lever pin travels in an arc with a radius, $a\ q = a\ n$, thus determining the rise of the dashpot plunger. For less rise a shorter radius would be used.

15. Drop-lever pin q should move equal distances on each side of horizontal center-line. The

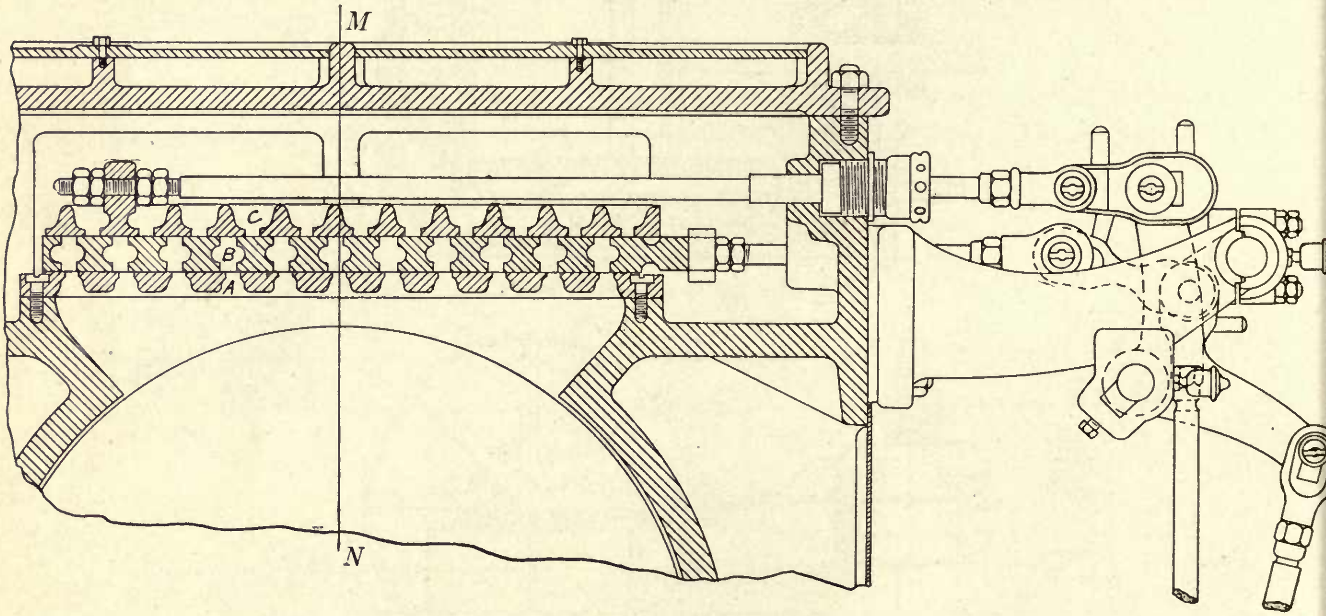


FIG. 77.—Section through $D\ E$ of FIG. 77a.

pins n and p on the rocker-arms, and the latch-pin q must all swing through the same length of arc $= n_2\ n_1$. Therefore lay off points q_1 and q_2 symmetrically about the center-line $a\ d$.

16. q_2 and q_1 correspond to extreme hook-latch positions. The distance between the hook latch and the pin on the rocker must be great enough to allow hub and arm length of hook to maintain latch effect, if desired, to end of swing. This distance is determined practically by the design of the hook, and in this problem the distance between q_2 and p_2 may be assumed to be great enough if an angle of 30° ($q_2\ a\ p_2$) is taken.

17. With the point p_2 determined, the angle between the two arms ($p_2\ a$ and $a\ n_2$) of the rocker is determined. Lay off the central and extreme forward positions of the arm $p_2\ a$.

18. Determine corresponding points for steam-valve on the crank end.

19. Lay off exhaust port, $t\ u = 1\frac{1}{8}$ inches; exhaust-lap, $u\ v = \frac{3}{32}$ inch, and exhaust port through valve, $1\frac{1}{8}$ inches.

Avoidance of Dead-Points in Valve-Gear Mechanism.

20. By trial-and-error adjust the length of the valve-arm *b x* (it is to be noted that the exhaust valve is never closed by a dashpot, and that it remains under the control of the wrist-plate all the time), and the link *x y*, so that *v* just over travels *t* to *z*, which gives smooth wearing effect. Neither *b x* and *x y*, nor *x y* and *y e* should cross a "dead-center" between their extreme positions. Locate central and extreme positions of the arms and links operating this valve.

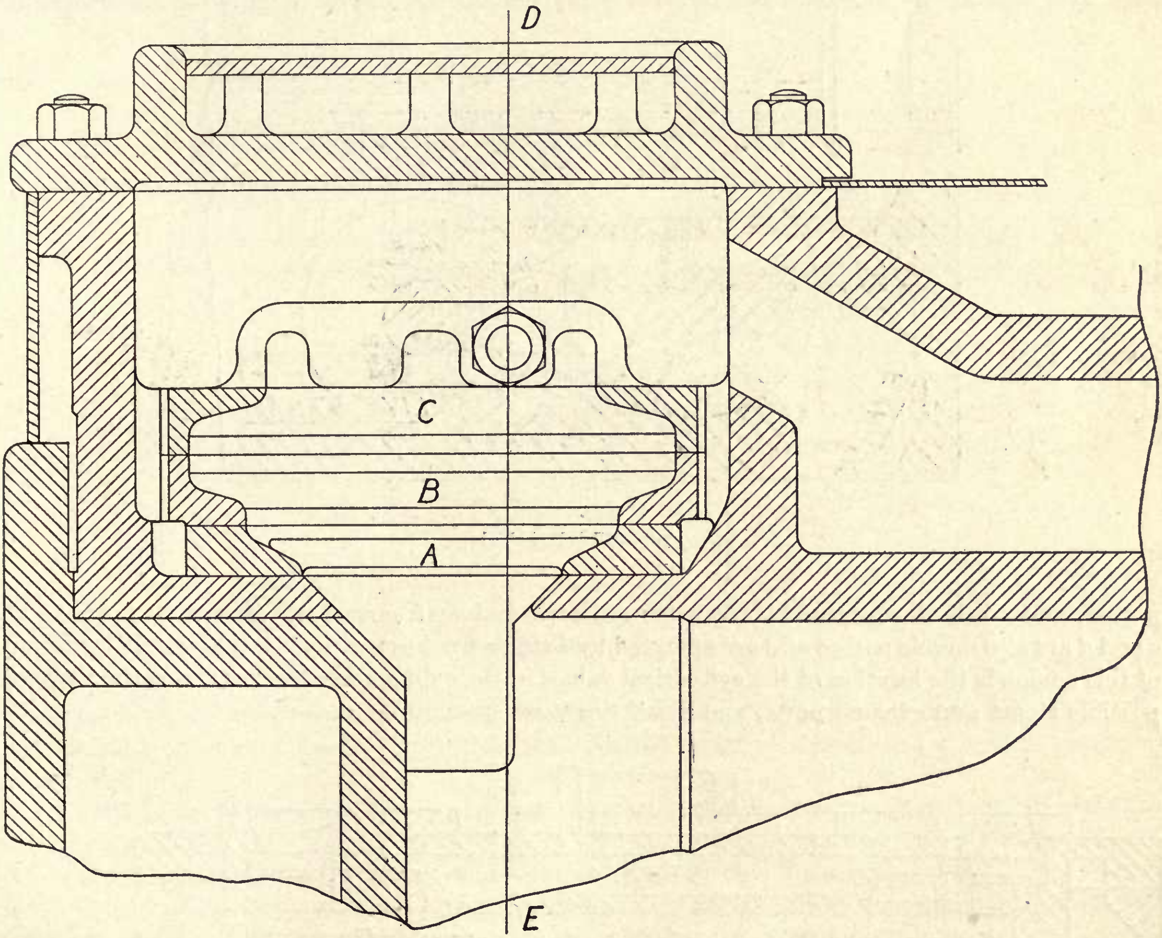


Fig. 77a.—Section through *M N* of Fig. 77.

21. Make the exhaust valve-arm at *c* the same length as that determined at *b*, and draw in the central and extreme positions.

22. In addition to showing the arms and links throughout for the three positions in characteristic line work, the passage-ways in the valves themselves must be indicated by the same character of line work, as per example at *b*. Place ¾-inch circles at all pin joints, except on the large rocker *f e g*.

23. Place dimensions at all points indicated in the sketch. Place the words "Forward" and "Return" in their proper places on the two arrows.

EXAMPLES OF PRACTICAL VALVE CONSTRUCTION.

The Corliss form of valve is used in a number of different makes of steam engines that are not classed as Corliss engines, an example being one of the types manufactured by the Atlas Engine Works and illustrated in Figs. 76 and 76a. The live steam valves are shown at *a* and *b*, are double

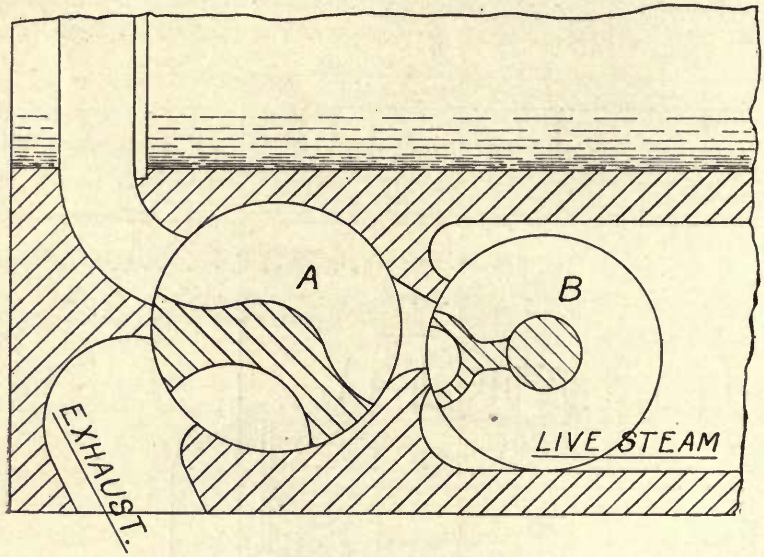


Fig. 78.—Wheelock Valves.

ported, and are both operated by the same eccentric and shaft-governor. The exhaust valves at *c* and *d* are also double ported and are operated by a single fixed eccentric. A characteristic feature of this engine is the location of the cylindrical valves in the cylinder heads, thus giving the shortest possible steam and exhaust ports, and small clearance space.

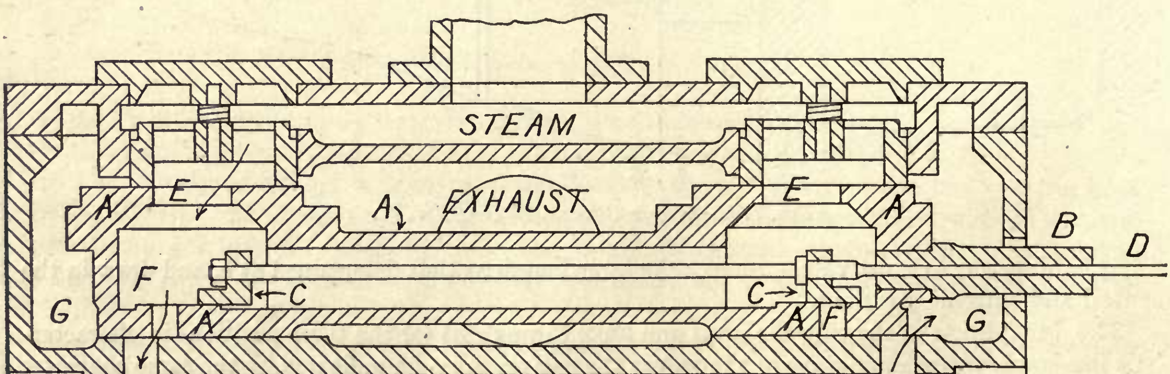


Fig. 79.—Buckeye Flat Valves.

A prominent valve of the Polonceau type, and combining the features of the double valve, and the four-part valve, is shown in Figs. 77 and 77a. It is also one of the pronounced types of the gridiron valve, and is used on the McIntosh, Seymour & Co's. engine. The section shown (Fig. 77) lies in a transverse plane (*D E* of Fig. 77a) close to the cylinder head. *A* is the valve-seat, *B* the main valve, and *C* the auxiliary or cut-off valve. There are two such valves, one at each end of the cylinder for the admission of live steam. In the two exhaust valves the auxiliary block is omitted. The advantage of the gridiron valve is small travel and friction, while it may be set to give small clearance and be operated by gearing, as shown in Fig. 77, to give practically no motion when idle,

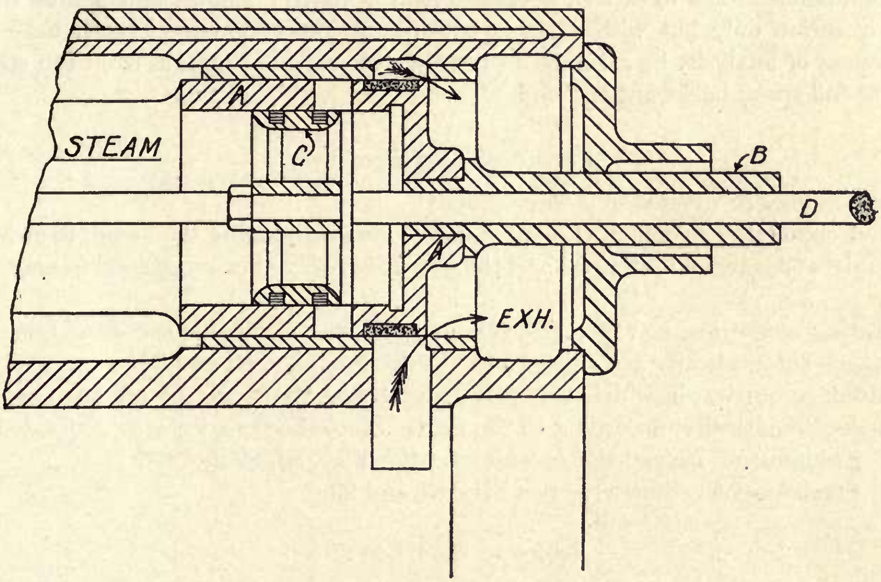


Fig. 80.—Buckeye Round Valve.

and to give cut-off at or near its maximum velocity. Its disadvantages are, delicate adjustment for lead owing to the numerous divisions of the port, limited range of cut-off, and expensive construction.

A valve of the Gonzenbach type is the one belonging to the Wheelock engine, shown in longitudinal section in Fig. 78. Although it has four independent parts, it belongs properly to the double-valve class, with one set of parts at each end of the cylinder. *A* is the main valve, which is designed to govern admission, release, and compression and *B* the auxiliary, which is designed to regulate cut-off only, and which is operated by a tripping mechanism under control of the governor.

A special valve of the Polonceau type is that used on the Buckeye engine, as shown in Fig. 79. The parts marked *A* form a steam-tight box, except at the opening *E* and the port *F*, and comprise the main valve. The blocks *C* form the auxiliary, or cut-off valve, which is operated through the rod *D* and a rotating eccentric, by the fly-wheel governor. *B* is a hollow valve-stem operating the main valve through a separate eccentric keyed permanently to the shaft. The two valve-stems *B* and *D* receive their motion through a compound rocker peculiar to this type of valve-gear. The main and cut-off valves each have uniform travel, and both are arranged so that cut-off takes place, whether early or late, when the cut-off valve is moving at or near its fastest rate.

Piston-valves mav also be used to give an independent cut-off, as shown in Fig. 80.

SECTION IV.—ECCENTRICS AND SHAFT-GOVERNORS.

Eccentrics.

An engine is "reversed" when its direction of running is changed from "over" to "under" or vice versa, and is usually accomplished by a special form of gear or link-mechanism between the valve and main shaft.

When the valve and eccentric are direct connected "reversal" can only be obtained by a wide movement of the eccentric. Eccentrics that move automatically by means of the centrifugal force due to weights in the fly-wheel, generally have a narrow motion and permit the engine to run in one direction only but with variable cut-off. In the eccentrics treated below, they will, for completeness of analysis, be assumed to have a wide range of motion from full speed forward or "over," to full speed backward or "under."

Classification of Eccentrics.

All eccentrics may be divided into three classes:

(1) Fixed eccentrics, in which the valve-travel always remains the same; likewise the angle of advance unless the engine is stopped and the eccentric rekeyed or refastened in another position. See Fig. 81.

(2) Rotating eccentrics, in which the valve-travel always remains the same but the angle of advance changes automatically according to the load. See Figs. 82 and 89.

(3) Slotted eccentrics, in which the valve-travel and the angle of advance both change at the same time automatically, according to the load. Slotted eccentrics are subdivided into,

(a) Swinging or curved-slot eccentrics. See Figs. 83, 87 and 88.

(b) Straight-slot eccentrics. See Figs. 84 and 93.

Reversing With Eccentrics.

To reverse an engine with any one of these eccentrics, it would be necessary to move the eccentric-center from c or d (Figs. 81–84) to f or g, as may be seen in Figs. 85 and 86, where the eccentric has been moved from $a\ c$ to $a\ g$. The arrows on the crank show the resultant change in the direction of running. In every case, Figs. 81 to 84, $l\ a\ c$ is the angle of advance and $a\ c$ the half valve-travel for one position of the eccentric, $l\ a\ d$ is the angle of advance and $a\ d$ the half-travel for another position of the eccentric, etc.

Exercises Showing the Relations Between Eccentric Positions and Zeuner Diagrams.

As essential exercises the student should here draw diagrams similar to Figs. 85 or 86, assuming sizes for all parts and angle of advance.

(1) For engine running over, with crank set at dead-center, crank end.

(2) For engine running over with straight reversing rocker-arm (see Fig. 25) when the crank is on dead-center, head end.

Other exercises necessary to a full understanding are:

(3) The drawing of Zeuner diagrams for head end only for the eccentric-center positions, c, d, e and g in Figs. 82–84. Draw these diagrams double size and assume the same steam- and exhaust-laps throughout, and note the effect on lead, and on the four principal crank positions.

Referring to Figs. 87–89, a is the shaft center, b the eccentric-center, $a\ b$ the half valve-travel, $e\ a\ b$ the angle of advance, and $a\ r$ the crank position in each case.

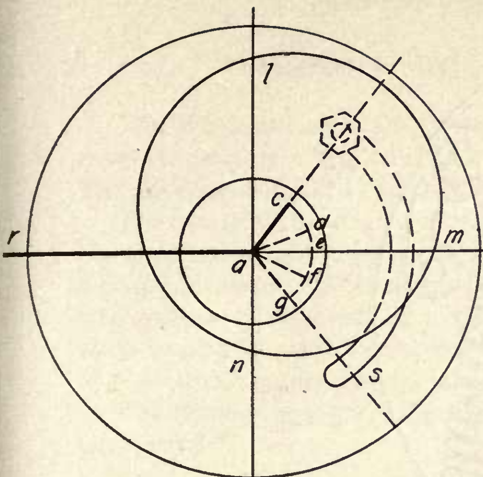


Fig. 81.—Fixed Eccentric.

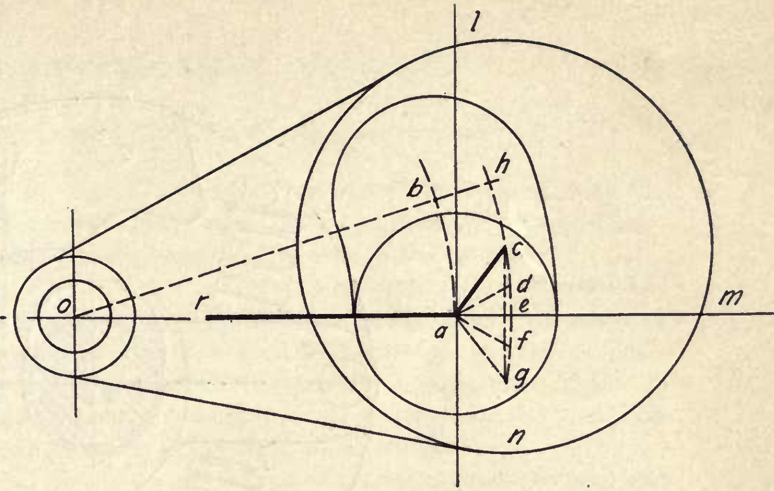


Fig. 83.—Swinging or Curved-Slot Eccentric.

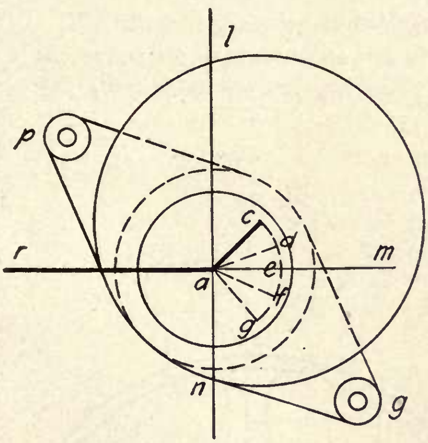


Fig. 82.—Rotating Eccentric.

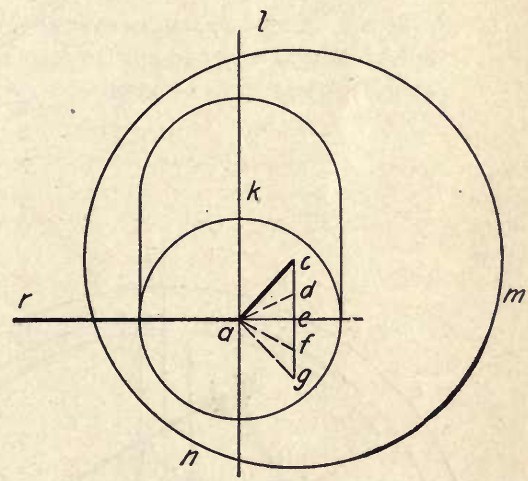


Fig. 84.—Straight-Slot Eccentric.

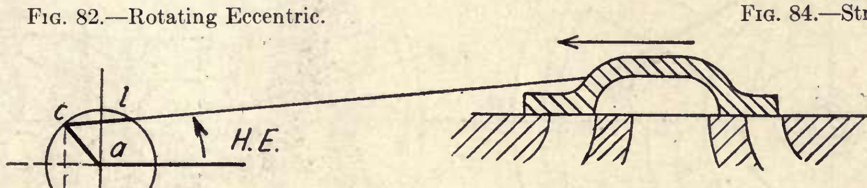


Fig. 85.—Running "Over."

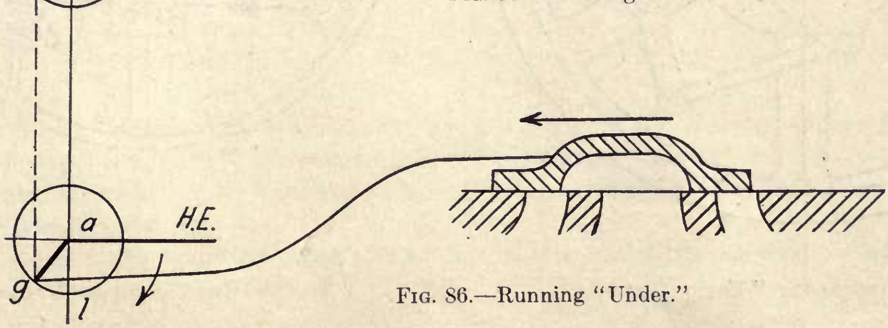


Fig. 86.—Running "Under."

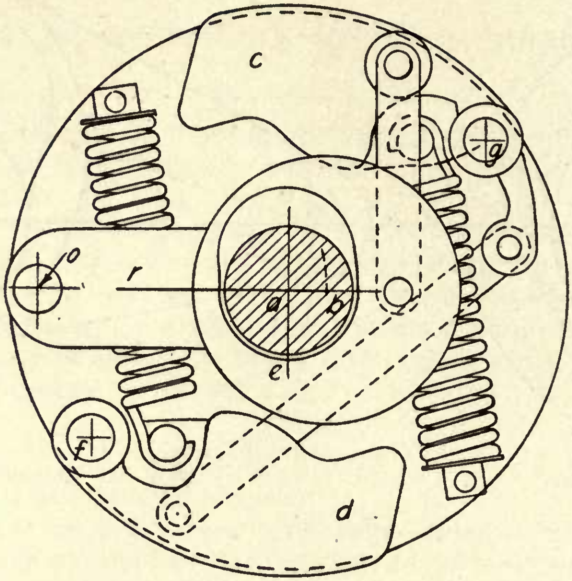


FIG. 87.—Westinghouse Shaft-Governor.

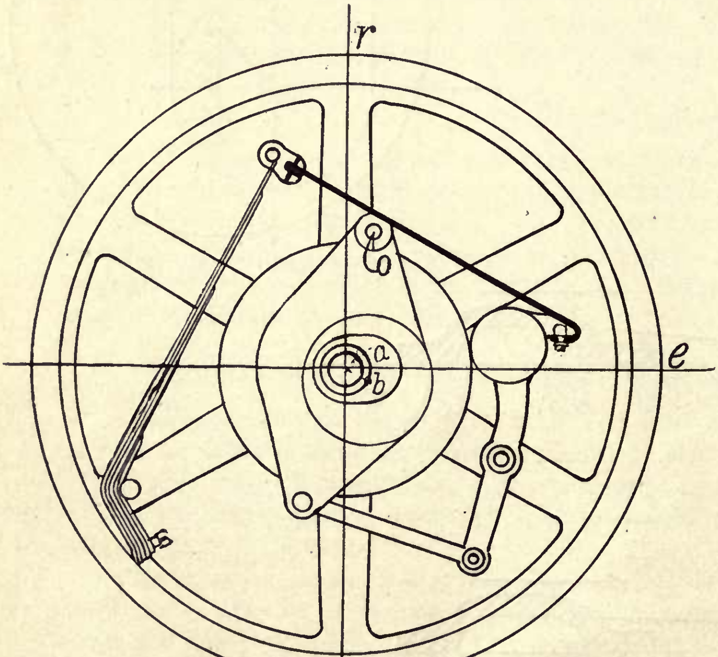


FIG. 88.—"Straight-Line" Engine Governor.

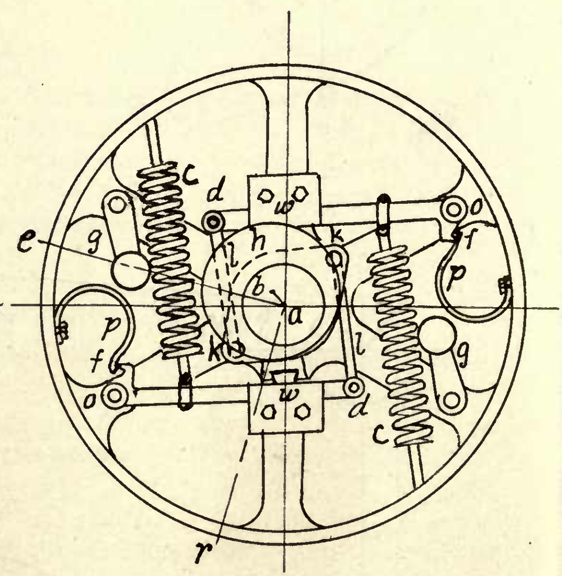


FIG. 89.—Buckeye Shaft-Governor.

Examples of Practical Eccentric and Governor Construction.

The Westinghouse shaft-governor, Fig. 87, has a curved-slot eccentric. It is shown with the governor weights *c* and *d* full out. The eccentric swings about the point *o* on the fly-wheel arm. The governor weights swing about the points *f* and *g* respectively, also on the fly-wheel.

The Straight-Line engine governor, Fig. 88, has also a curved-slot eccentric. It is shown with the governor weight in, and as the engine speeds up the eccentricity is reduced and the angle of advance increased. This eccentric is swung about a pivot, *o*, specially located so as to give a positive lead for ¾ to ⅞ cut-off and a negative lead at the shorter cut-offs. The purpose of this is to neutralize to some extent the results of too early compression and release on short cut-offs and too little compression on late cut-off, all of which follow from shifting eccentrics.

The Buckeye governor, Fig. 89, shows a rotating eccentric with minimum angle of advance and latest cut-off.

Effect of Location of Pivot in Curved-Slot Eccentrics.

In using the curved-slot or swinging-eccentric, it makes a difference which side of the shaft the eccentric is pivoted on, as Fig. 90 will show. The solid part of this figure is similar to Fig. 83. The dotted lines are added to represent the eccentric if it were designed to swing about *r* instead

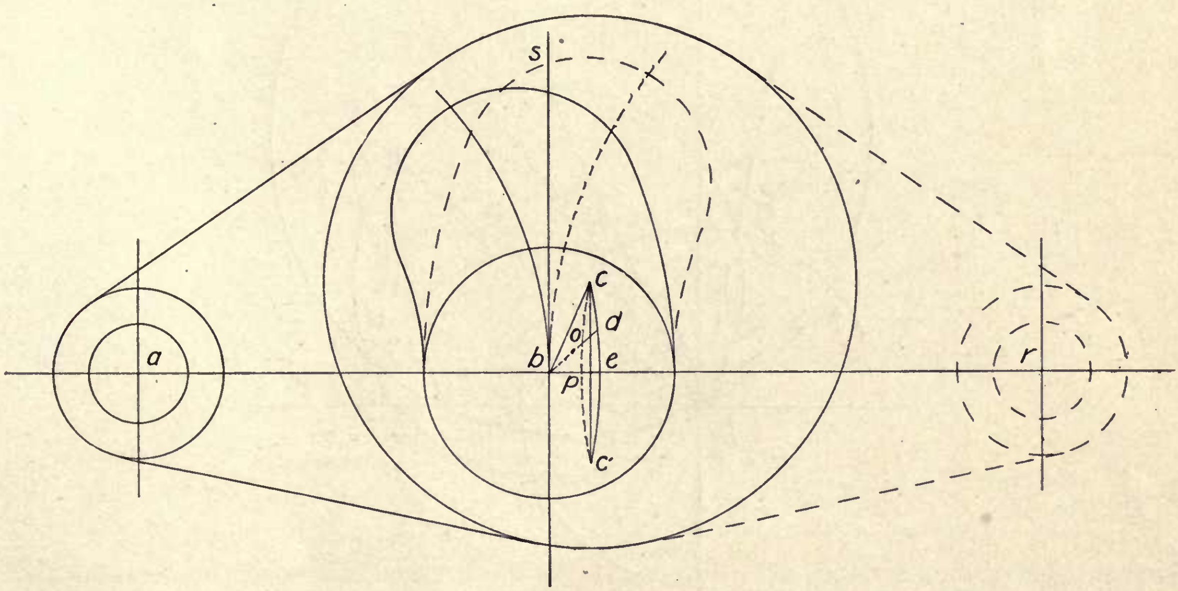


Fig. 90.

of *a*. *b* is the center of the main shaft. *c* is the center of the eccentric-sheave and, for the position shown, *s b c* is the angle of advance, and *b c* the eccentricity.

If the eccentric is moved about *a* so as to give the angular advance *s b d*, *c* will go to *d* and *b d* will be the eccentricity.

If the eccentric is moved about *r* so as to give the same angular advance *s b d*, *c* will go to *o* and *b o* will be the eccentricity.

When *c* reaches *e* the engine is in mid-gear, and if *c* were moved to *c′* the engine would be in full-gear, running in reverse direction. By constructing a series of Zeuner circles with the angles of advance *s b c*, *s b d*, and *s b e* (see Figs. 91 and 92), it may be shown that the lead *increases* largely toward mid-gear by using pivot *a*, and that it is *reduced* toward mid-gear by using pivot *r*. It should be noticed that while the lead increases in one case and decreases in the other, the *angle of lead* increases in both, but at a much slower rate in Fig. 92.

The forms of shaft-governors already illustrated show:

1. The eccentric which turns on the shaft, giving always the same valve-travel but varying angles of advance, Fig. 89.

2. The swinging- or curved-slot eccentric, Figs. 87 and 88.

The following illustrations show:

3. The straight-slot eccentric, Figs. 93 and 94, and the equivalent of the straight-slot eccentric, Figs. 95–97, which illustrate the Armington and Sims governor.

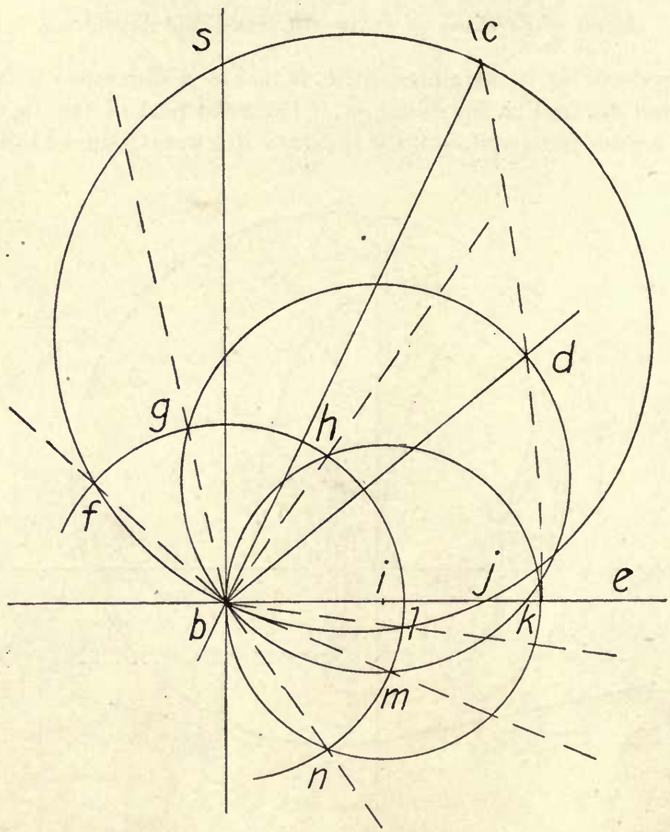


Fig. 91.

4. The swinging pivot, which takes the place of the swinging eccentric, Fig. 98, illustrating the American-Ball engine. The complete engine is shown in perspective outline in this case, so that the student may form a comprehensive idea of a complete set of connections from the governor mechanism to the valve. In all the other governor illustrations here given the eccentric-rod *k* screws into, or is bolted to, the eccentric-strap which works on the eccentric-sheave.

Explanations of Figs. 93 to 98 are as follows:

In Fig. 93 the frame *c c* is securely attached to the shaft whose center is *o*. The position shown is for the engine at rest; as it speeds up the weights *w w* fly out, and the center of the eccentric-sheave *a* moves straight across and changes the angle of advance, for example, from *d o a* to *d o a'*, and the eccentricity from *o a* to *o a'*, preserving constant lead.

Another form of straight-slot eccentric is that manufactured by the Fitchburg Steam Engine Co., illustrated in Fig. 94. The two pins, *s* and *e*, in the eccentric-strap move in short arcs following closely the vertical center-line, thus giving to the center, *b*, of the sheave approximately the same straight-line motion as is secured by the Watt's parallel motion mechanism. Thus practically constant lead is obtained. The weights *o o* balance the weights of the eccentric and its strap, and also the valve and valve-rods in vertical engines, taking the effort to move these parts off the governor. Both the centrifugal weights act in unison through the links, *k c*, *k f*, and lever arms, *s d*, *e d*, to move the eccentric as the engine changes speed. The engine may be made to govern when running in a reverse direction by transferring the ends of the connecting links, *k c*, *k f*, from *c* to *f* and

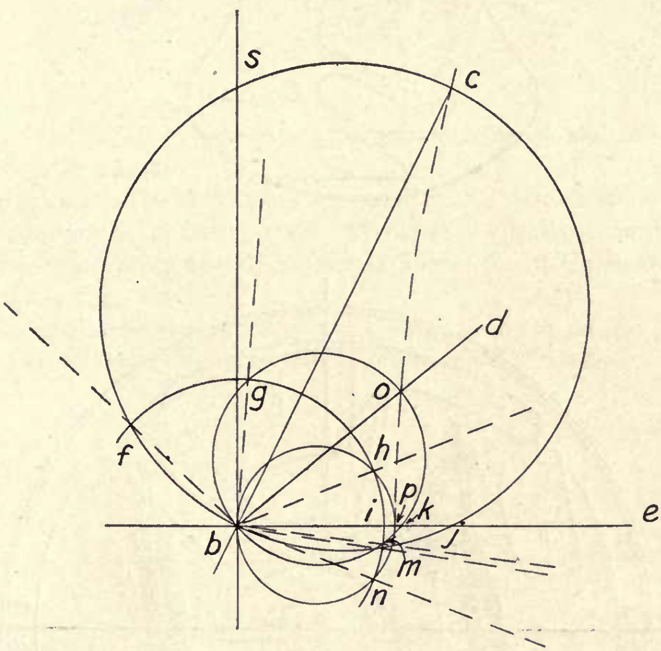


FIG. 92.

from *f* to *c* respectively, and putting on a new eccentric-sheave. The governor, in the position drawn in Fig. 94, is at rest or running at very low speed, *r a* being the position of the crank, *e a b* the angle of advance, and *a b* the half valve-travel.

The Armington and Sims governor, Figs. 95–97, has a combination of two eccentrics, one being loose on the shaft and the other surrounding it. The inner eccentric is connected, through arms and links, to weights *w* and *z* as shown; the outer eccentric is connected to *w* only. The weights *w* and *z* are pivoted to the arms *d* and *m* of the fly-wheel, which is keyed to the shaft *s*. Fig. 95 shows the governor mechanism when the engine is at rest, and Fig. 96 running at top speed. In the latter position the governor has a minimum eccentricity, as shown at *s' r'* in Fig. 96, and also in the

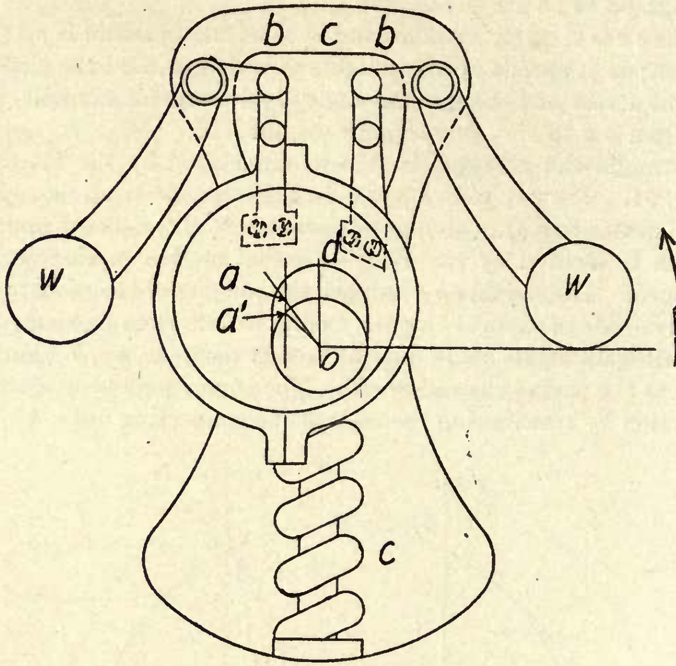


FIG. 93.

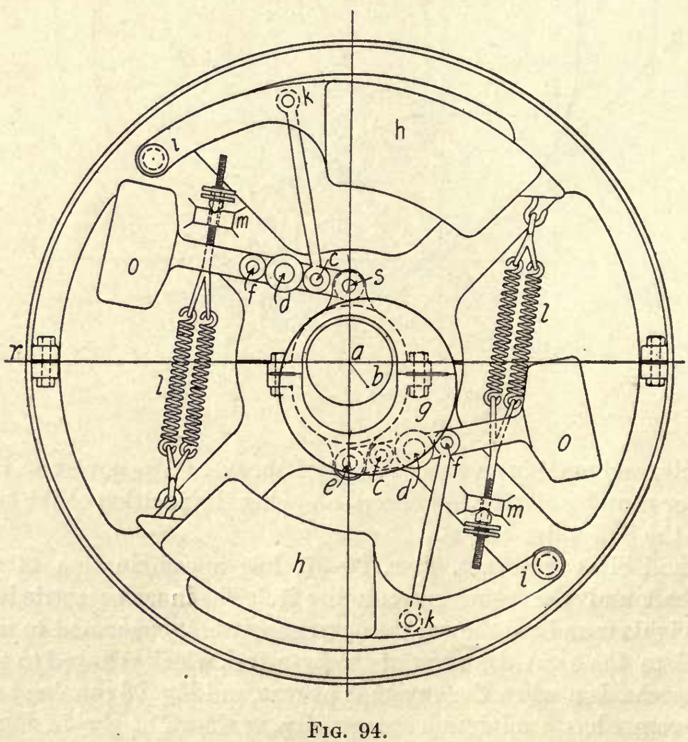


FIG. 94.

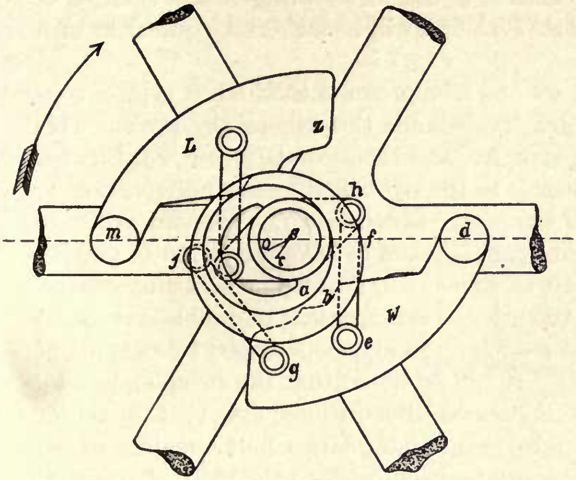


FIG. 95.

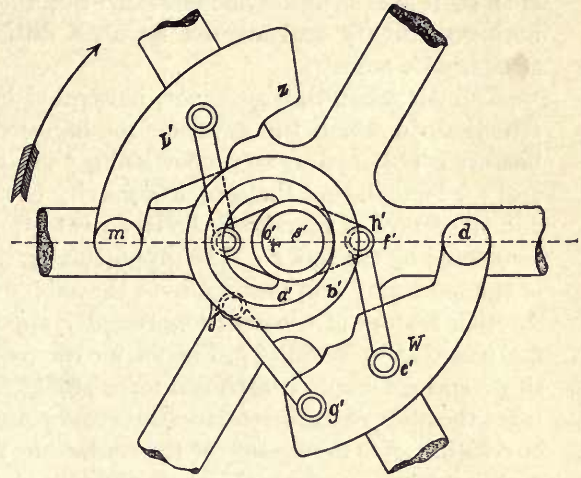


FIG. 96.

center-line sketch, Fig. 97, in which *s* represents the center of the shaft, and the other letters, the corresponding points of Figs. 95 and 96.

When starting up, the eccentricity of the inner eccentric is *s o* (see Figs. 95 and 97), that of the outer eccentric is *o r* with respect to the inner eccentric, and the effective eccentricity of the combination is *s r*, the angle of advance being *t s r*. The proportions of the mechanism are such that the theoretical angle *j o r* remains con tant.

As the speed of the engine increases, the points *e*, *g*, *j*, *o*, *r* and *h* go to *e'*, *g'*, *j'*, *o'*, *r'* and *h'* respectively and *s r'* becomes the eccentricity and *t s r'* the angle of advance. If the path of *r* (*r r'*)

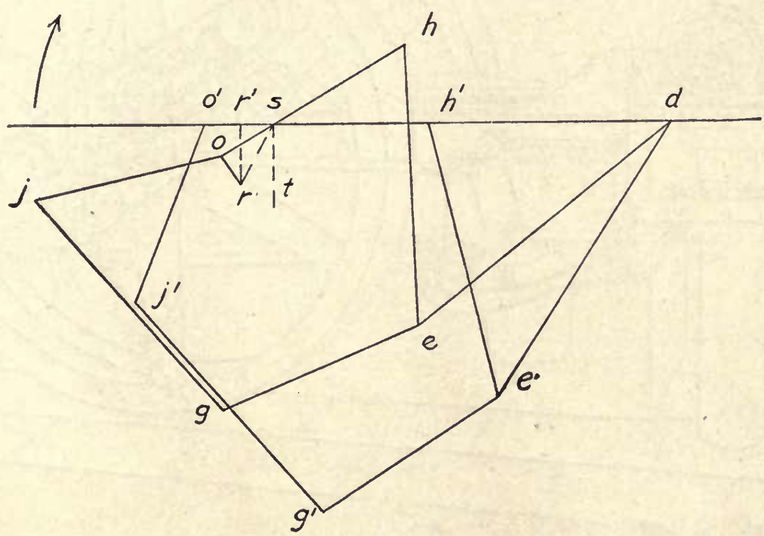


FIG. 97.

is at right-angles to the line of stroke *s d*, this combination of eccentrics will give the changes in both eccentricity and angular advance with a *constant lead*, and will be an exact equivalent of a straight-slot eccentric.

The American-Ball governor, illustrated in Figs. 98 and 99 represents their latest type of construction, in which the governor mechanism is in gravity balance throughout the cycle. This balance is obtained by so proportioning the governor arm, *b c*, and the secondary arm, *r f*, Fig. 98, and *j f*, Fig. 99, that the center of gravity of the former is to the right of the center of rotation *a*, and the center of gravity of the latter is to the left of the center of rotation *f*. The two arms are connected by the link *d e*. In proportioning these arms care is taken to have the center of gravity of the secondary arm practically at the axis of the shaft so that it will develop no centrifugal force. Another feature of construction, recently applied, is the use and arrangement of double springs, as illustrated at *m n* and *m p*, Fig. 98, for the purpose of reducing the ill effects caused by swaying of single springs due to centrifugal force and gravitation. It will be noted that the swinging pivot *g* takes the place of the eccentric-sheave and strap shown in previous illustrations; also, that the center of rotation, *f*, is to one side of the center-line for the purpose of giving larger port openings at the earlier cut-offs. *a o g* is the angle of advance, *k o* the crank position, and *o g* the half-valve travel; *o h* is the half-valve travel at minimum cut-off.

Another governor, simple in construction and similar in action to that shown in dotted Fig. 90, and giving a diagram quite similar to that shown in Fig. 92, is manufactured by the New York Engine Co., at Watertown, N. Y. It is illustrated diagrammatically in Fig. 100, *k o* being the crank, *n o g* the angle of advance, *g o* the half valve-travel, *o h* the lap, and *o l* the lap plus lead. The weighted

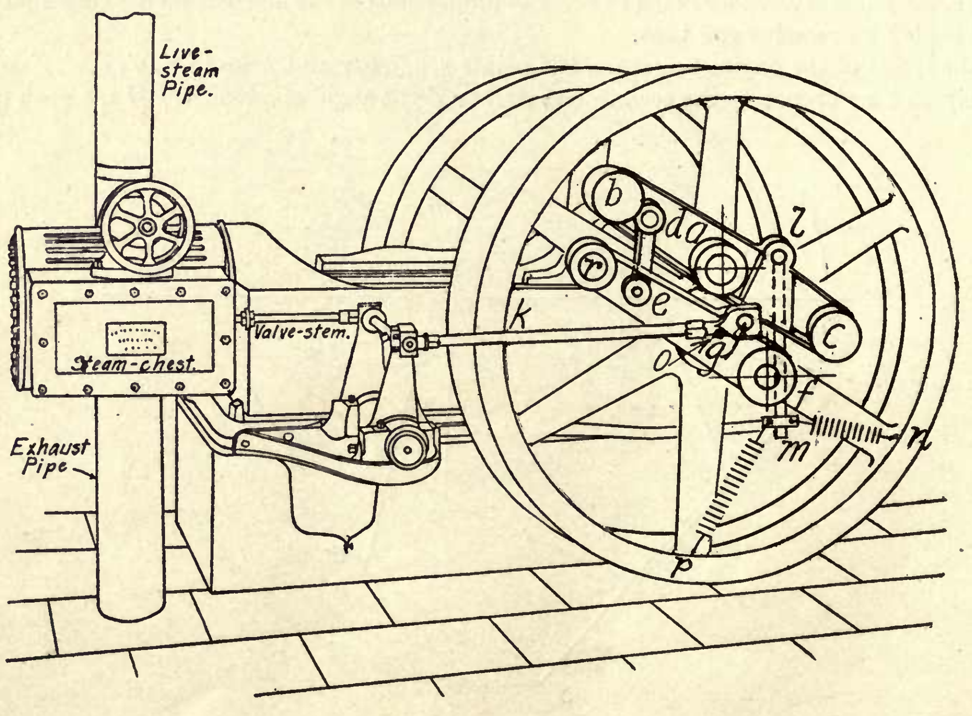


FIG. 98.

bar, *b b*, swings on the pivot *f* which is secured to the fly-wheel arm; and its position is controlled by the speed of the engine and counteracting spring *r s*.

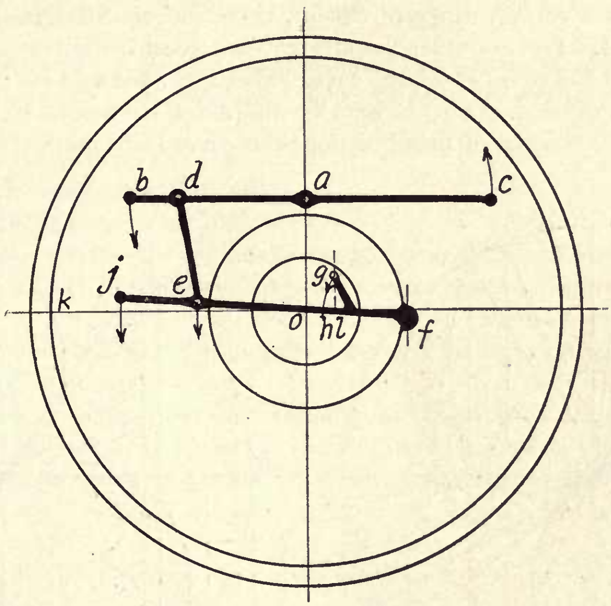


FIG. 99.

Comparative Indicator Cards From Different Kinds of Eccentrics.

Comparative results obtained on the indicator card by the different forms of shaft-governors are illustrated in two of the most used forms in Figs. 101 and 102. It will be seen, in general, that the compression, w', x', y', increases very rapidly and becomes very large with the earlier cut-offs; also

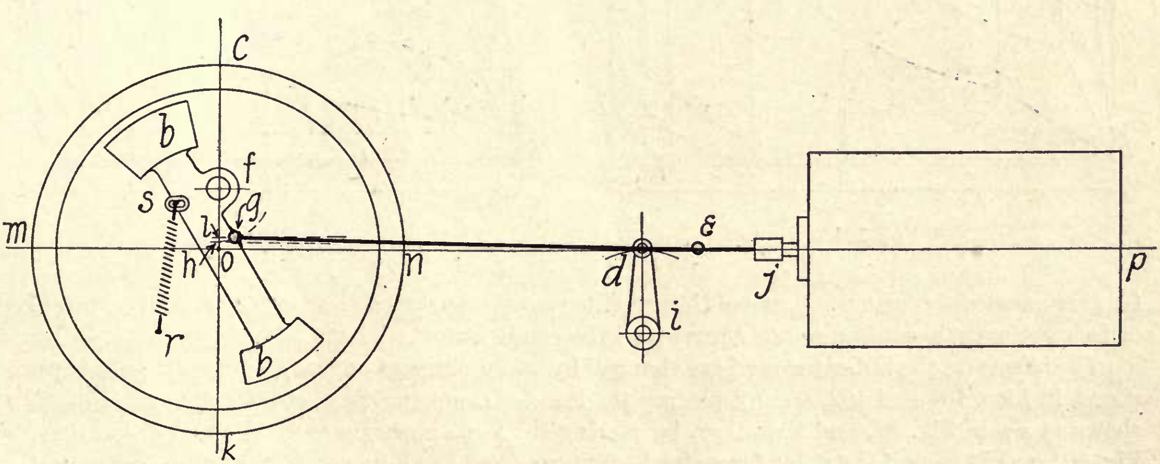


FIG. 100.

that the preadmission, n', o', p', increases rapidly in Fig. 101. The release also comes too early with the early cut-off and the compression too late with the late cut-off for smoothest and most economical

running. Negative exhaust lap, which is used under some conditions, has the effect of making the compression much later on the earlier cut-offs, and the release earlier.

Different makes of shaft-governors are designed to meet certain average conditions, and to give best action throughout a certain range of cut-off, these different designs leading to the claims of different manufacturers. For example, the straight-slot eccentric gives a wider range of cut-off, 0.228 to 0.780 against 0.332 to 0.792 for the curved-slot, Figs. 101 and 102; the curved-slot, a more uniform maximum compression, as may be seen by comparing the points, n', o', p' in the two figures; the straight-slot a smaller range of preadmission; the curved-slot more area in the card, etc., etc.,

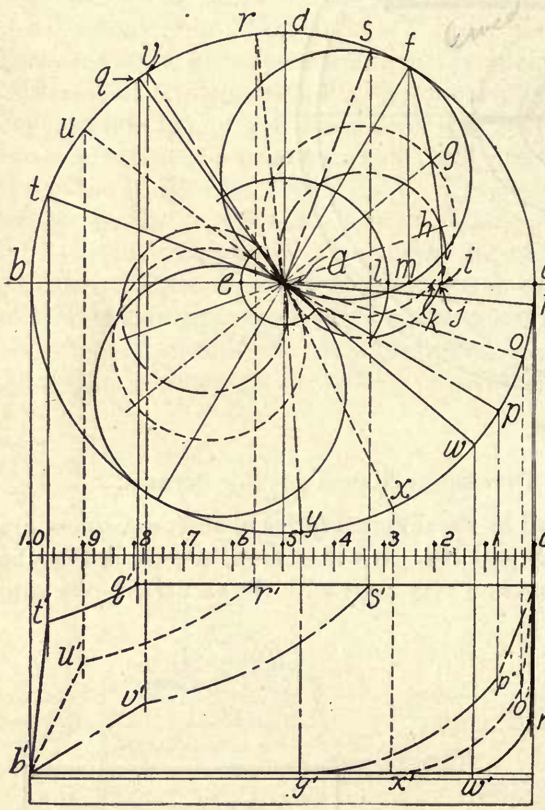


FIG. 101.

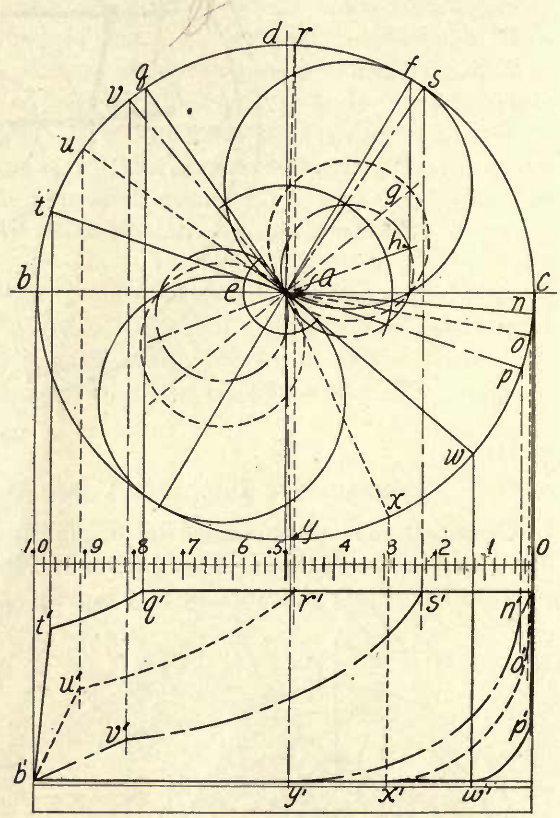


FIG. 102.

for given angles of advance. Some of these points may be construed as advantages or disadvantages, or as lesser evils, according as one sums up all the conditions of a design.

The effects on the indicator card are changed by two prominent engine manufacturers from those shown in Figs. 101 and 102, one by placing the locus of the eccentric center for different cut-offs as shown at *c p* in Fig. 92, and the other, by placing the locus approximately in the position that *f i*, Fig. 101, would have if the point *f* remained stationary and *i* were swung to the left in an arc until it met the horizontal center-line *a c*.

Since preadmission is, under ordinary conditions, the most powerful factor in "compression," or smooth running over the dead-centers, it must be looked to critically in design work. It will be seen

from Figs. 101 and 102 that preadmission is always variable, even when the lead is constant, and that the straight-slot eccentric gives the narrower range of preadmission.

Shaft-Governors.

A paper read by Mr. Frank H. Ball before the American Society of Mechanical Engineers (see Transactions, A.S.M.E., Vol. XVIII, page 290), shows three forces available in shaft-governor construction, as follows:—

1. Centrifugal force (Fig. 103).
2. Tangential accelerating force (Fig. 104).
3. Angular accelerating force (Fig. 105).

Each of the Figs. 103, 104 and 105 represents the governor wheel, or fly-wheel to which the governing mechanism is attached. *a* is the shaft, *b* is the mass, *c* the pivot supporting the mass through the link *d*.

The rotation of the wheel in Fig. 103 will cause the mass *b* to move outward and produce centrifugal force.

In Fig. 104 the mass *b*, being connected radially to the pivot *c*, can have no motion due to centrifugal force. On account of its inertia it will, however, move relatively in the direction shown by the arrow *e* if the angular velocity of the wheel increases, or in the direction *f* if the angular

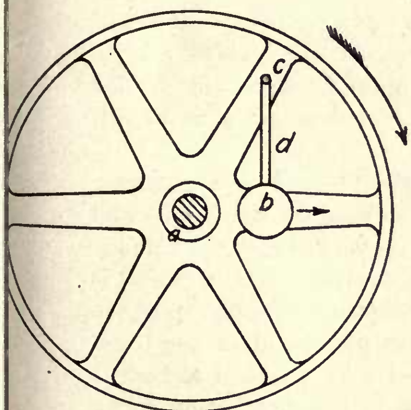


Fig. 103.

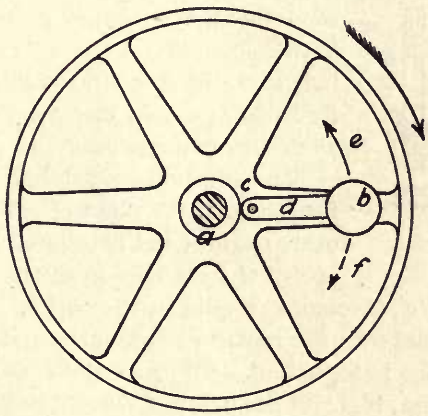


Fig. 104.

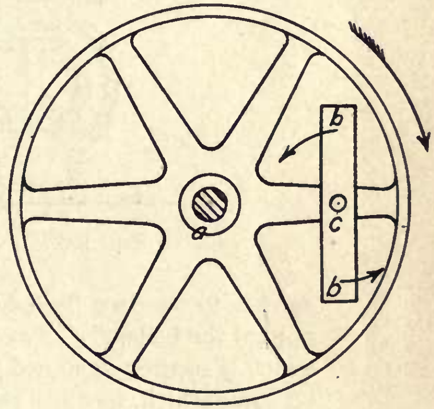


Fig. 105.

velocity of the wheel decreases. (Governors using this force—or the following one, or both—are sometimes termed "inertia" governors, but owing to the forces 1, 2 and 3 being combined in so many different ways by manufacturers, we will not speak of inertia governors as a class.) This force is called "tangential accelerating force."

Angular accelerating force, which is described "as the effect of the angular acceleration of the mass about its own center of gravity," is represented in Fig. 105, in which the mass *b* is distributed.

Effects Produced by Rate of Rotation and by Rate of Change of Rotation.

To distinguish further between centrifugal force and tangential accelerating force (commonly called "inertia"), it should be noted that the former depends on the *rate of rotation* only, while the latter depends entirely on the *rate of change of rotation*.

The "American Electrician," in the early part of 1902, illustrated twenty-two different forms, or makes, of the shaft-governor, all making use of centrifugal force, or centrifugal force in combination with tangential and angular-accelerating forces.

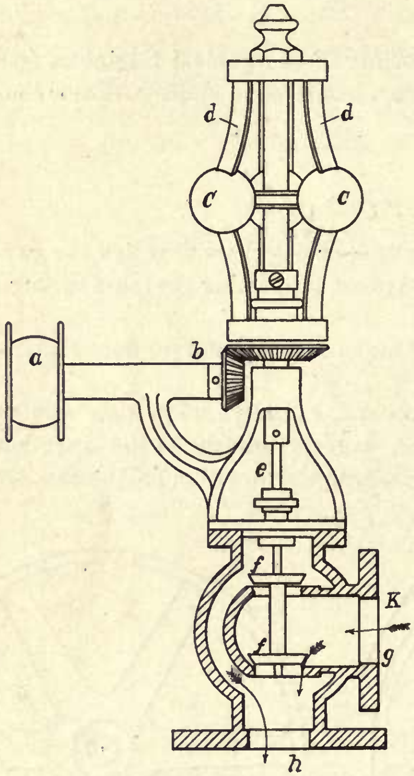


Fig. 106.

Throttling Governors.

The shaft-governor, as has been shown, regulates the speed of the engine by changing the point of cut-off. All governors which regulate in this manner may be placed in one class, and called "automatic cut-off governors." There is another class of governors which regulate the speed of the engine by throttling, or, in other words, by reducing or increasing the steam pressure while the cut-off remains constant. An example of this latter type is shown in Fig. 106, which is an illustration of the Pickering governor. A belt from the engine shaft drives the pulley, *a*, and this rotation is carried through the bevel wheels, *b b*, to the three weights, *c c c*, which are attached to the flat springs, *d d d*. Should the engine get above normal speed the weights, *c c c*, would fly out by centrifugal force, and in so doing would draw down the valves, *f f*, through the spindle, *e*, and so reduce the passage-way for live steam and consequently the steam pressure. *g* is live-steam inlet, and *h* the opening to the steam chest. The valves, *f f*, are balanced, the steam pressure being on all sides alike. The fly-balls, *c c c*, and the spindle, *e*, constitute what is known as the revolving pendulum.

The revolving pendulum is also applied as a governor for changing the point of cut-off, as used on the Corliss and other engines and illustrated in Fig. 69, in which *a* is a pulley operated by a belt from the engine shaft. The rotation is carried to the two fly-balls, c' c', through a spindle in the post *b*. As the engine speeds up above normal the balls, c' c', fly out and turn the rocker *e f* through a rod in the post *b* and an arm (e' *d*) which is securely fastened to the rock-shaft *d*. The governor-rods *g* and *h* are attached to knock-off cams which, by their rotation, regulate the point of cut-off, as shown in the explanation of the Corliss valve-gear.

Drafting-Table Problem, No. 6—Comparing Results From Straight-Slot and Rotating Eccentrics.

Construction of Comparative Indicator Cards: One Obtained from an Eccentric having a Straight Slot, which gives a Constant Lead with a Variable Travel and Angle of Advance; the Other Obtained by Rotating an Ordinary Eccentric, which gives a constant travel with a Variable Lead and Angle of Advance.

A comparison of the results obtained by the use of the two kinds of eccentrics mentioned in this problem may be shown to best advantage by assuming a concrete example in which both eccentrics are designed to cut-off at a given point in the stroke, and then moving each so as to produce cut-off at an earlier point in the stroke.

To facilitate the work the student may refer to the Zeuner diagram of Problem 1, in which a plain slide-valve was designed to obtain cut-off as given above. Construct the indicator card

for the head end for this case, using a live-steam pressure of 80 lbs. gauge, with 2 lbs. back pressure and a 40 spring. Assume a clearance volume of 5 per cent.

In Fig. 107 the dotted construction work is taken from the Zeuner diagram of Problem 1, *C O* representing the crank position for the given cut-off, *D K L* the steam-lap circle, *G F* the lead, and *D E F* the Zeuner circle.

To construct the Zeuner diagram for the rotating eccentric for any other cut-off, such as at *O J*, draw *K M* perpendicular to *O J*, and the arc *N E P* with *O E* as a radius. The intersection

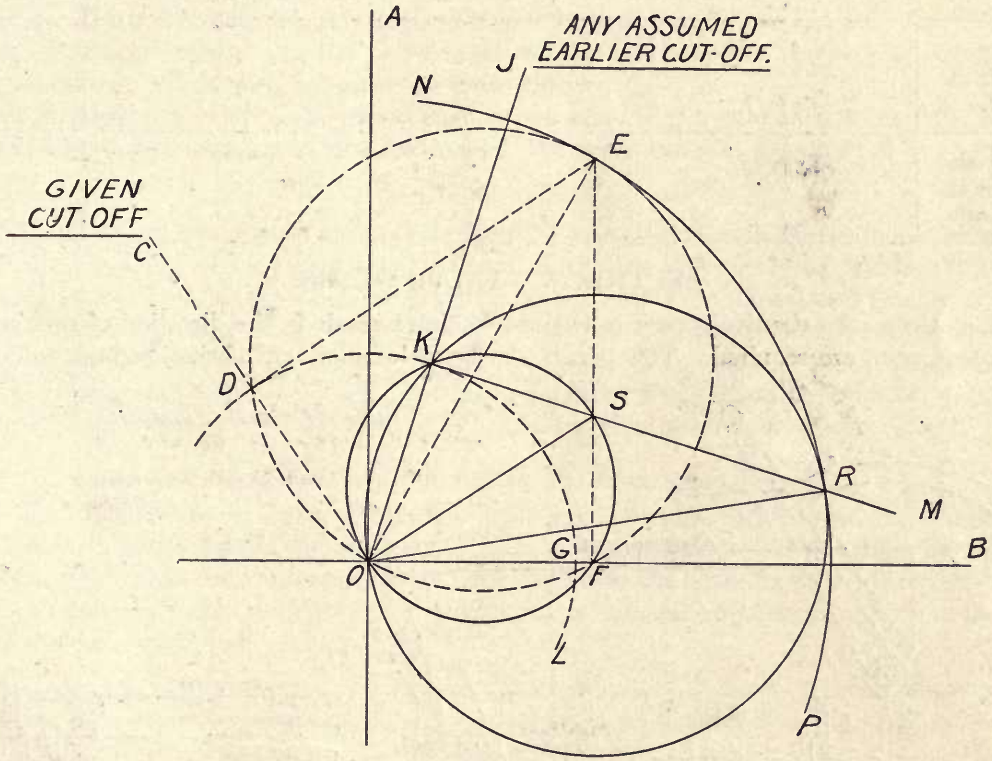


FIG. 107.

of *K M* with this arc gives the point *R*, the extremity of the diameter of the Zeuner circle for cut-off at *O J*.

To obtain the Zeuner diagram for the straight-slot eccentric for cut-off at *O J*, for example, locate the point *S* at the intersection of *K M* with *E F*, *E F being the line of constant lead.* Then *O S* equals the diameter of the Zeuner circle for the straight-slot eccentric cutting off at *O J*.

In the diagram for this problem, draw in the symmetrically situated Zeuner circles for the complete revolution, and locate and designate the piston positions for all events of the stroke. Enter the results in a table as follows:

Method of governing	Angle of advance	Travel of valve	Lead	Per Cent. of Stroke Completed When			
				Admission begins	Cut-off takes place	Release begins	Compression begins
Both eccentrics at...... cut-off							
Rotating eccentric at ... *C O.*							
Straight-slot eccentric at ... cut-off.							

SECTION V.—VALVE-GEARS.

Link-motions are extensively used in engines where reversals in the direction of running, variable speeds, etc., are required. This occurs chiefly in locomotives, marine engines, rolling mill

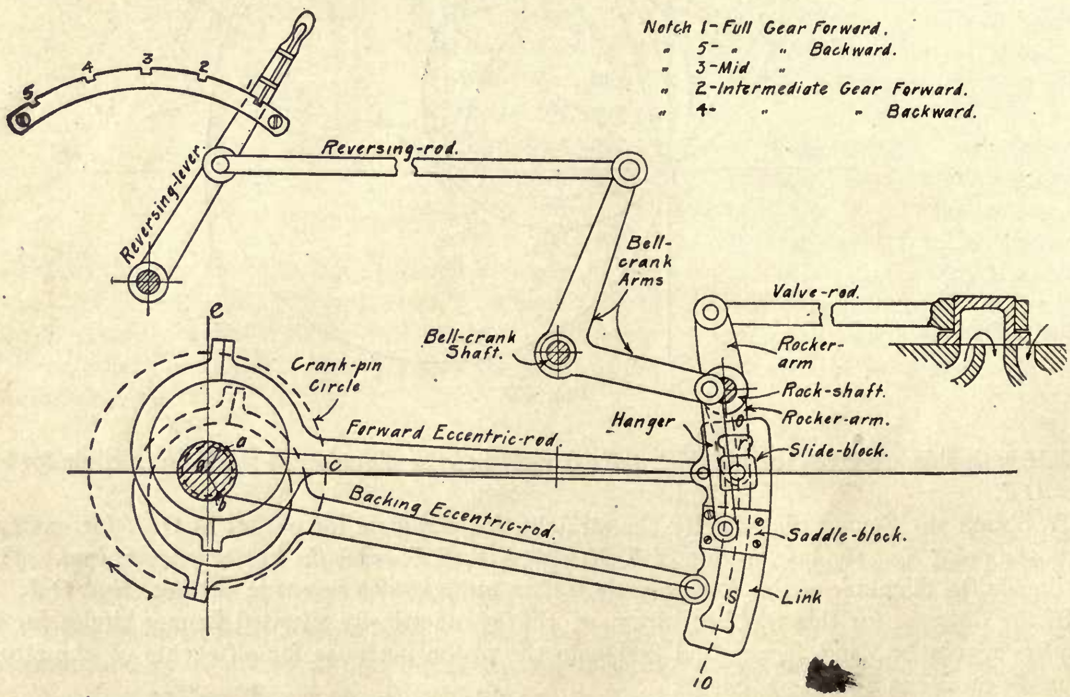


FIG. 108.

engines, etc. Perhaps the most largely used of all the link-motions is the Stephenson, shown in **Fig. 108**, which represents, in a diagrammatic manner, its application to the locomotive.

Stephenson Gear.

The explanation of the action, in a general way, is as follows: For the position shown, the valve is being operated, through the rocker, by means of the forward eccentric only. This is evident from the fact that the slide-block, which is pivoted to the end of the rocker-arm, is in direct line with the forward eccentric-rod; in this position the backing eccentric-rod has no influence on the valve motion.

If, now, the reversing-lever be moved so as to clasp in notch 2, the link, through the reversing-rod, bell-crank and hanger, will be raised so that the saddle-block pin is closer to the slide-block pin. The slide-block, which will then have a motion due to both eccentric-rods, will have a shorter horizontal swing, and the valve consequently will have less travel, less port-opening, earlier cut-off, and will furnish less power to the engine.

With the reversing lever in its central position, or notch 3, the saddle-block pin and the slide-block pin will be over each other, and the travel of the valve and port opening will be a minimum.

Method of Reversing.

The link and the connections are so designed that when the reversing-lever is moved to notch 5 the backing eccentric-rod is raised so as to be in line with the slide-block pin, which is thus drawn to one side or the other, except when the engine is on dead-center. This causes the rocker to turn on its shaft, and move the valve to the right or left to such an extent that the opposite port may open to steam and reverse the engine. When the engine is on dead-center the slide-block and valve will remain nearly stationary when the link is raised or lowered and the cylinder on the opposite side of the locomotive with its crank at 90° must be relied upon to start up.

A Valve-Gear at any One Setting Equivalent to an Eccentric.

The link, operated by two fixed eccentrics, is for any one phase or notch setting, a mechanical equivalent for a single curved-slot eccentric. Such an equivalent eccentric is sometimes called a virtual eccentric. The link, however, has an advantage over the latter, in that it is capable of such adjustment that practically nullifies the irregularity of cut-off and exhaust closure due to the angularity of the connecting-rod.

A more definite idea of the complex action of the link and its connecting mechanisms is shown in a graphical manner by the method given by Auchincloss, as illustrated in Figs. 109, 110 and 111. Fig. 109 is a center-line reproduction of Fig. 108, the line *o s r o* of the template, in Fig. 109, corresponding to the center-line *o s r o* of the link in Fig. 108.

In Fig. 110 the lines 00, 11, 22, etc., represent the successive positions of the center-line *o s r o* of the link during one cycle. The curves *u v* and *w x*, which are envelopes of the link center-lines of Fig. 110, are reproduced for clearness in Fig. 111. The length of the horizontal line included between these two curves measures the limits of the horizontal travel of the link. For example, if the slide-block pin is at *h* the valve-travel is *y z*, and the travel of the saddle-block pin is *p t*.

Detail Construction.

Briefly, the order of construction for the illustrations thus far referred to is as follows, keeping in mind that all that is shown in Figs. 109, 110 and 111 should be developed step by step on one single figure. The figures are separated here to avoid complication of line work.

In Fig. 109 take o' for the shaft center and $o'\ c$ for the crank on dead-center. Lay off points *a* and *b* (according to the angle of advance) as the eccentric-center positions for crank at $o'\ c$, and draw the eccentric-center circle. With the length of the eccentric-rods (*a m* and *b n*) and the dimensions of

the link template (*m n*, *m r* and *n s*) given, the eccentric-rods and template may be drawn in position as shown for the engine on dead-center. Then taking the lap plus lead from the angle of advance at *a*, lay it off from the template edge at *r* to *h*. From *h* draw the vertical line and on it lay off the given distance from *h* to *g* (the rock-shaft center). The line *h g* is then the position of the rocker

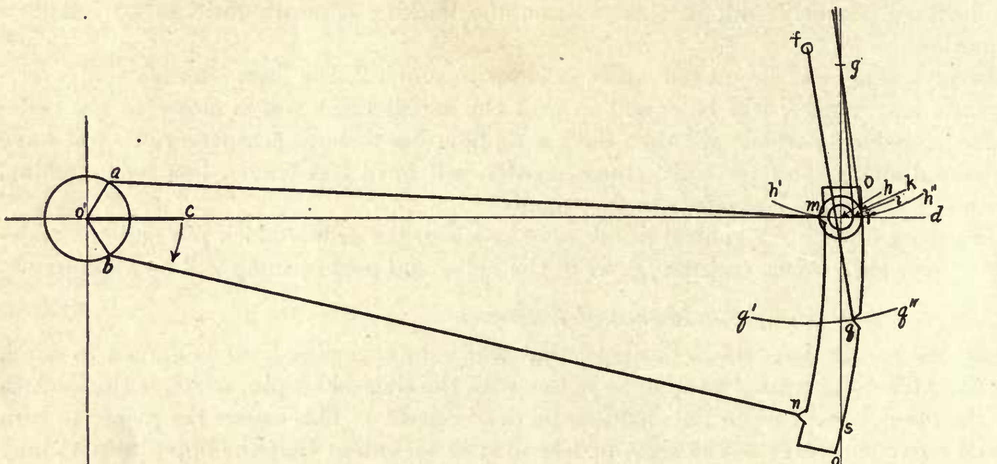


Fig. 109.

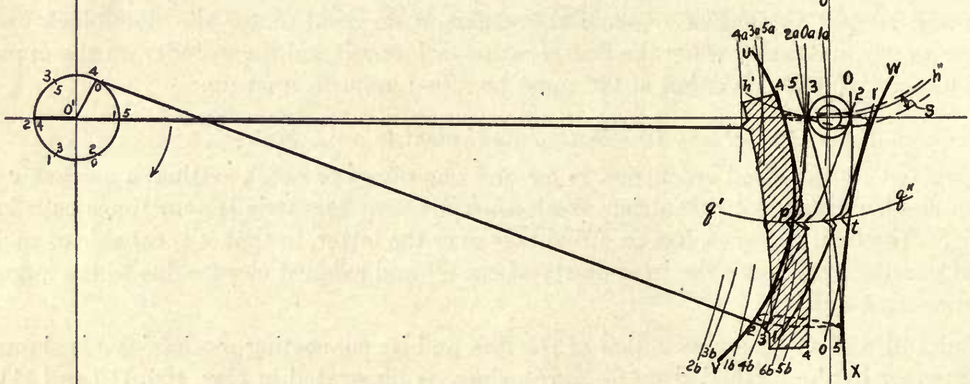


Fig. 110.

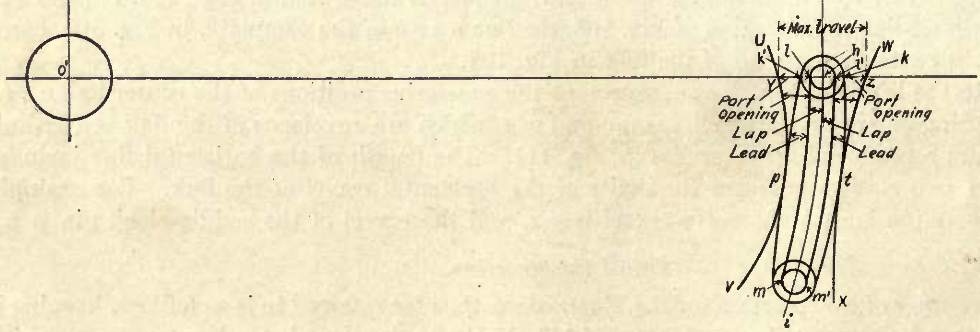


Fig. 111.

when the valve is central, and *g r* is the position of the rocker-arm after the valve has moved off center a distance equal to the lap plus the lead. (This position is represented in Fig. 108, where the small port-opening shown represents the lead. It will be noted that the crank *o c* is on dead-center, as it should be for this position of the valve.)

Lay off $r\,l$ = lead; then $h\,l$ is the lap. For illustration, assume the lap to be ⅔ (lap + lead). Then with the center-line of the rocker-arm at $g\,l$ the valve will have moved a distance off center equal only to the lap, and admission will have begun.

The next step in the layout of a problem by this method consists in cutting a template of some suitable material, with the center-line $o\,r\,s\,o$ of the link as its right-hand bounding line, and with notches at m, n and q corresponding to the two eccentric-rod pin points and the saddle-block pin point. This latter point must always lie on the arc $q'\,q\,q''$ described about f as a center, and the points m and n must always lie on arcs described with the eccentric-rods as radii in their successive positions. The exact location of f depends on frame-work construction, and in this problem may be assumed approximately as shown. The lines on which m and n must always be found are obtained as follows: Divide the eccentric circle, Fig. 110, into a convenient and sufficient number of parts depending upon the accuracy required—six are shown in this case. These divisions are laid off in the direction of rotation, first from a, for the forward eccentric, and then from b for the backing eccentric. From each of these division points, draw the arcs, $1a$, $2a$, and $1b$, $2b$, etc., using the eccentric-rod length as a radius. These arcs will contain the points m and n of the template in its successive posi-

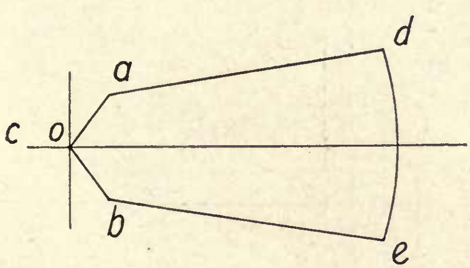


Fig. 112.

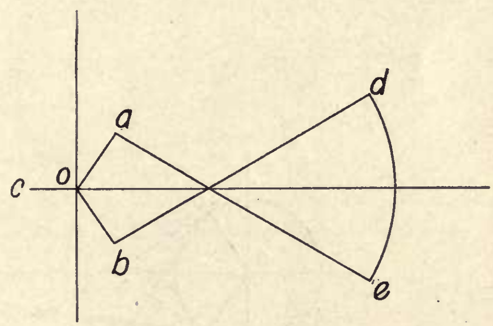


Fig. 113.

tions and in addition the point q of the template must always lie on the arc $q'\,q\,q''$. The template may now be adjusted for the six positions, thus giving the lines 0 0, 1 1, 2 2, 3 3, 4 4 and 5 5 in Fig. 110. Draw envelopes to these curves as shown at $w\,x$ and $u\,v$.

In Fig. 111 these envelopes are reproduced. Draw the arc $h\,i$ with a radius $o'\,h$. With h as a center draw the lap, and lap + lead circles, with $h\,l$ and $h\,k$ as radii, respectively. Draw similar circles at m. Then with o' as a center draw the arcs $l\,m$ and $l'\,m'$; the travel of the slide-block pin, from the center arc $h\,i$ to $l\,m$ or to $l'\,m'$ is, approximately, just sufficient to take up the outside lap of the valve. Draw a curve tangent to the lap + lead circles at the top and bottom, and also tangent to the envelopes at p and t; the travel of the slide-block pin between the two curves $h\,i$ and $k\,t$ is just sufficient to move the valve a distance equal to the lap plus the lead. It will be seen that the lead will vary with the different elevations of the link. The distance from $l'\,m'$ to $w\,x$ is the port-opening.

"*Slip.*"

In the practical adjustment of the link and its connecting mechanism for precise work, one great difficulty arises on account of the "slip" which occurs between the slide-block and the link. This slip is shown in Fig. 110 as follows:

The upper dotted loop shows the path of the point r on the surface of the link during one revolu-

tion of the crank. A corresponding point of the surface of the slide-block can only travel in the arc $h' h h''$ about g. It will be seen therefore that slip during one revolution equals the distance s.

In planning a link motion it is necessary to reduce this slip as much as possible, on account of wear. When the link is in such a position that the saddle-pin is directly over the slide-block pin, the slip is comparatively small.

Open and Crossed Rods.

Links may be connected with the eccentrics in two different ways, either by "open rods," as shown in Fig. 112, or by "closed" or "crossed" rods, as shown in Fig. 113. When the centers of the two eccentrics (a and b) lie between the shaft and the link, and the projections of the rods do not intersect, the rods are said to be "open." When the eccentric-centers lie between the shaft and the link, and the projection of the rods cross each other, the rods are said to be "crossed." It should be noted that the position of the crank has nothing whatever to do with open or crossed rods.

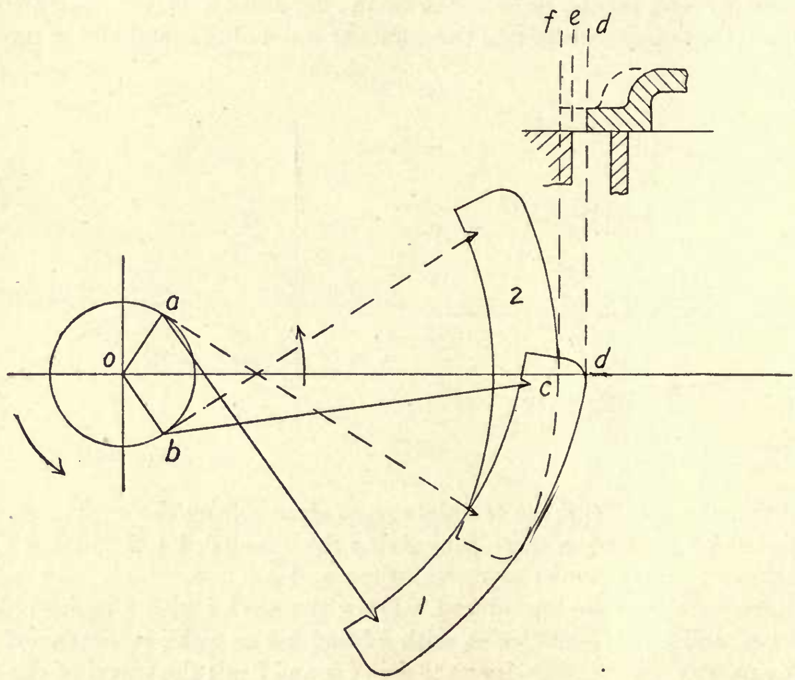


Fig. 114.

Other things being equal, open and crossed eccentric-rods give quite different steam distributions. In Fig. 114 the link is shown in position 1 in full-gear, and in position 2 in mid-gear. It will be seen that the lead for full-gear is $d\,e$, and that for mid-gear it is $-e\,f$. In other words, the lead decreases from full to mid-gear, even to negative lead sometimes, as shown in this case, with crossed rods. With open rods the lead increases from full to mid-gear, being shown equal to $j\,k$ for full, and equal to $j\,l$ for mid-gear, in Fig. 115. The effect of short open rods is to increase the lead more rapidly, as also shown in Fig. 115, where $m\,p$ is greater than $j\,l$.

Inasmuch as the half-travel of the valve in mid-gear is equal only to the lap + lead (Fig. 111), and if the lead for mid-gear is zero, or a minus quantity (as represented in Fig. 114), it will be observed

that the half-travel of the valve is equal to, or less than, the lap of the valve in crossed rods; in which cases steam will not be admitted to the cylinder. Therefore, in a crossed-rod design where the lead at full-gear is small enough, it is possible to shut off steam by placing the link in mid-gear. With open-rods this cannot be done, no matter how small the full-gear lead for the reason that, as has been stated, the lead increases from full to mid-gear, and therefore the steam-ports are open a definite amount for every setting of the link when the engine is in dead-center. In practice, generally, open-rods are used, and the lead for mid-gear position ranges from $\frac{1}{4}$ inch to $\frac{1}{2}$ inch with a common value $\frac{3}{8}$ inch, while the full-gear lead ranges from $\frac{1}{16}$ inch to $\frac{3}{16}$ inch, governed principally by the length of the eccentric-rod.

Several practical considerations in connection with link mechanism should be pointed out:

1. That the bell-crank shaft must be situated a sufficient distance above or below the center-line of motion so that the eccentric-rods do not strike it when raised or lowered to full-gear.

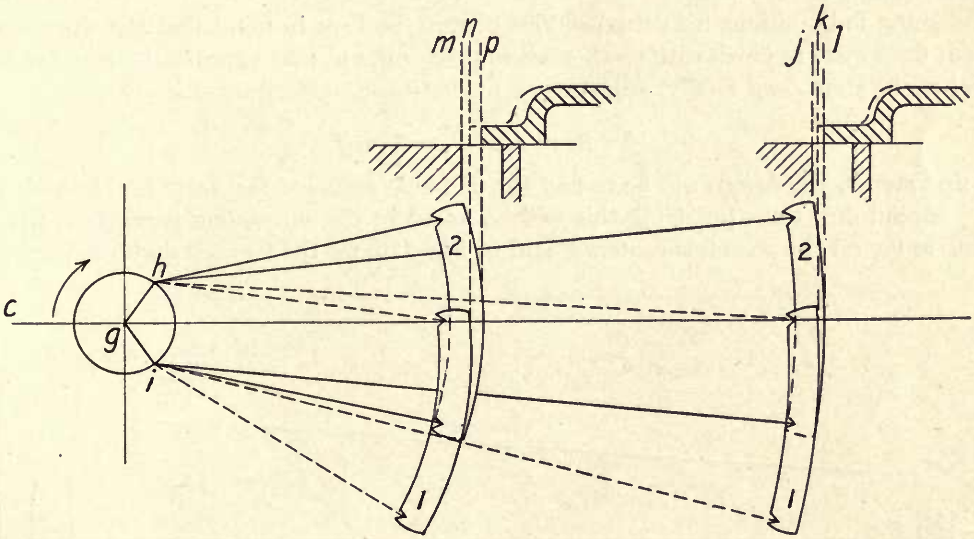


FIG. 115.

2. The hanger should be of such length that the link will not conflict with the bell-crank in any position. The length of the bell-crank arm is usually equal to, or greater than, the hanger.

Relation Between Center-Lines of Valve-Gear and Engine Cylinder.

3. So long as the angular advance of the eccentric is laid off from a line at right angles to the central line of the link-motion, the latter may be arranged to any inclination to the piston motion without affecting the action of the link. In Fig. 108 the center-lines of the link-motion and the piston-motion coincide. With proper mechanical connections the valve motion would remain the same if the link, eccentric-rods, and eccentric-sheaves were considered rigid with respect to each other while they were turned through any desired angle about O as a center, the center-line of the engine remaining fixed.

Design of a Stephenson Gear.

In order to make a direct and practical application of the method involved in laying out a link-motion for an actual case, let the following data be given:

Ratio of crank to connecting-rod = 1: 7½.
Eccentric circle diameter = 5½ inches.
Maximum cut-off = 0.92 stroke.
Center to center of eccentric-pins (*m* to *n* of Fig. 109) = 13 inches.
Center of eccentric-pin back of link-arc (*m k* and *n s*, Fig. 109) = 3 inches.
Mid-gear lead = ⅜ inch.
Length of hanger = 18 inches.
Exercise Problem: Find the steam-lap, full-gear lead, point of suspension of link, and location of tumbling-shaft.

The mid-gear lead is given in this problem for the reason that the valve-travel is least at mid-gear, as may be seen in Figs. 110 or 111, and the lead therefore constitutes a much larger percentage of the steam-port opening near mid-gear than it does in full-gear. In fact at mid-gear the lead and steam-port opening are the same.

In designing link-motions for actual service it must be kept in mind that the work cannot be carried out from start to finish with mathematical precision, but that approximations must be made in several of the steps, and finally, adjustments made to secure the desired results.

To Find Mid-Gear Travel.

The first step in the design will be to find the mid-gear travel of the valve for the assigned conditions. Special directions for doing this will be found in the succeeding paragraph, the general plan being to lay off the eccentric-centers *F* and *B*, Fig. 116, for the forward and backing eccentrics

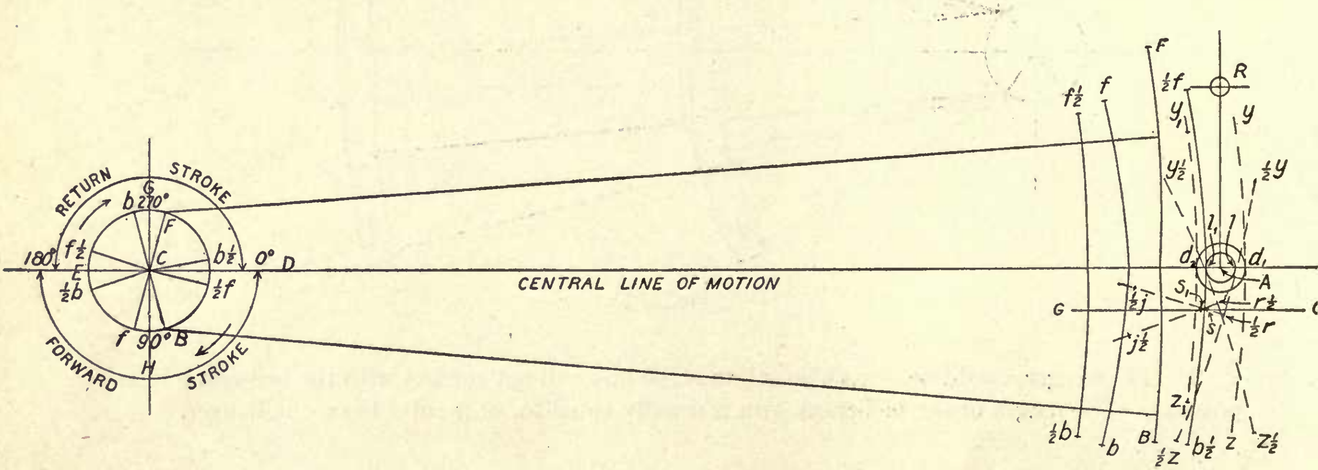


Fig. 116. Fig. 117.

(for a 92% cut-off) when the piston is at one end of the stroke, and *f* and *b* likewise when the piston is at the other end of the stroke. Then, taking the eccentric-rod lengths and a template (such as in Fig. 118) whose edge coincides with the center-line curve of the link, the extreme positions of the link-arcs *y z* and y_1 z_1, Fig. 117, may be found by placing the eccentric-centers at *F* and *B*, and *f* and *b*, respectively, thus giving d_1 d_2 as the travel in mid-gear.

In following this plan the student may need to refer to the following directions as to detail: Figs. 116 and 117 are connected, the distance from the point *F* in Fig. 116 to the arc *F* in Fig. 117

being equal to the length of the eccentric-rod, which is 46¼ inches (49¼″ – 3″). The angle *G C F* is the angle of advance for 92% cut-off, and is found to be equal to about 17° for this problem.*

The student should note and understand that on account of the reversing-rocker which is used

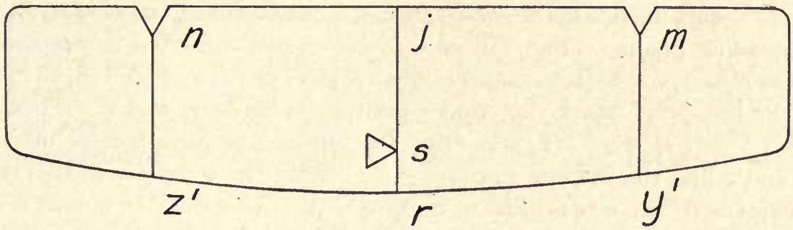


FIG. 118.

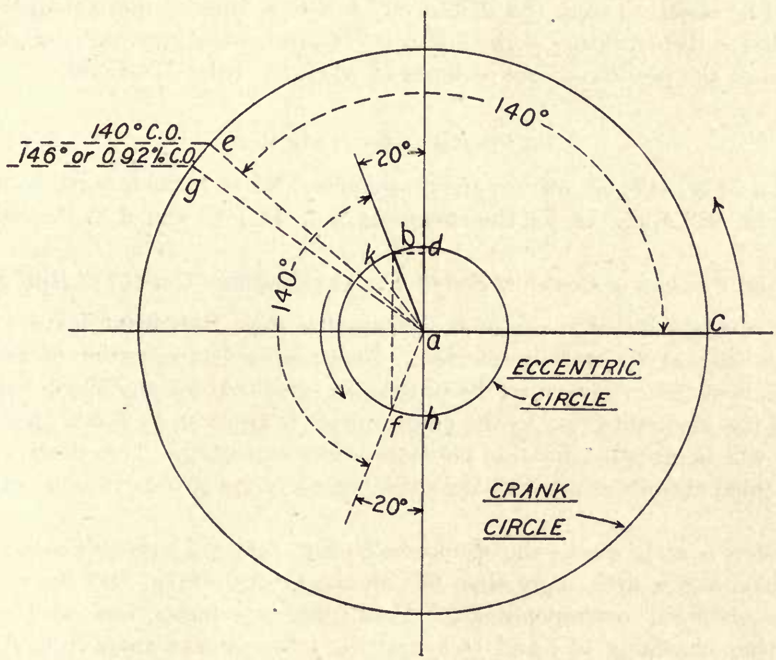


FIG. 119.

with the Stephenson link, the eccentric must follow instead of precede the crank, and that the actual angle of advance must be laid off back of 90° instead of in advance of it.

*The reason for connecting 17° angular advance with 92% cut-off may be explained by the following independent example according to explanation on pages 3 and 4: Given the crank *a c*, Fig. 119, and the eccentric *a b* with an angle of advance of 20°. If we neglect lead, and give the valve a steam-lap equal to *b d*, the steam-port will just be opening at the crank position shown. When the eccentric *a b* gets to *a f*, which also makes an angle of 20° with *a h*, the steam-port is just closing and the eccentric has traveled 140°. The eccentric and crank being rigidly connected, the latter has also traveled 140° when cut-off takes place. If, therefore, the position of the crank at cut-off is given in a problem, as 140°, for example, it may be shown that the angle of advance necessary to secure this equals ½ (180° — 140°) = 20°, if lead is neglected. If the lead angle is taken into account, and is, say, 10°, as shown at *b a k*, Fig. 119, then the cut-off will occur 10° sooner with the same lap, or at 130°, as may be seen by a study of Fig. 119. There being no reversing-rocker in this explanation, the eccentric precedes the crank, and the angle of advance is laid off ahead of 90°.

As the whole design is approximate, and the full-gear lead smaller than ⅜" but not yet definitely known, it may be neglected in laying out the eccentric positions F and B in Fig. 116 (for the crank at C D) and f and b (for the crank at C E)

Since the eccentric-rod pins are 3 inches back of the link-arc, the eccentric-rod itself is 49¼ inches − 3 inches or 46¼ inches long. With this as a radius strike the arcs F, B, f, b, of Fig. 117, with the corresponding points of Fig. 116 as centers. Then construct a template having a link-arc of 49¼ inches radius, and with incisions as shown at m and n of Fig. 118, 13 inches apart and 3 inches back of the link-arc. Mark the point j midway between m and n. Then adjust the template on the arcs F and B, Fig. 112, and draw the link-arc position y z for mid-gear. Similarly with the arcs f and b find the link-arc position y_1 z_1. With the template in this position draw j r, Fig. 118, coincident with the central line of motion C A. Also draw m y' and n z' parallel to j r. The arcs y z and y_1 z_1 cross the center-line of motion at d_1 and d_2, which therefore measure the valve-travel at mid-gear. The point A, midway between d_1 and d_2 determines the location of the rocker-shaft. It should be observed that the distance C A is now slightly more than the assigned value of 49¼ inches, due to the curvature of the link-arc. This increase, however, is neglected in practice, and A R is taken as the position of the rocker-arm when the valve is central.

To Find the Lap of the Valve.

From d_1 and d_2, Fig. 117, lay off the given mid-gear lead of ⅜ inch equal to d_1 l and d_2 l_1, and draw the circles l l_1 and d_1 d_2. A l is the steam-lap, l d_1 the lead and A d_1 the half-travel for mid-gear.

To Find Position of Center of Saddle-Pin for Equalized Cut-Off at Half Stroke.

The location of the saddle-pin center is perhaps the most important feature in the design of a link-motion, and should be carefully selected. By a proper determination of its location the cut-off on the two ends of the cylinder may be practically equalized for any single point in the stroke.

Inasmuch as the inequality due to the crank angles is greatest at about ½ stroke, the irregularity of cut-off will be greatest for this position unless corrected. This design will therefore be laid out to give equal cut-offs at one-half the stroke on each end of the cylinder, with a symmetrical valve.

When the piston is at ½ stroke the crank-pin D, Fig. 116, will have advanced a little less than 90° forward stroke, and a little more than 90° on the return stroke, and the eccentric-centers F and B will have advanced correspondingly. Find these two angles, and lay the forward one off from F and B, thus obtaining ½ f and ½ b, and the return-stroke angle from f and b which will give the points f ½ and b ½. With the four points just found, as centers, and with a radius equal to the eccentric-rod length (46¼ inches), describe the arcs ½ f, ½ b, f ½ and b ½, Fig. 117. Adjust the template with m and n on the arcs ½ f and ½ b, respectively, and with the link-arc passing through l. This will give the position (½ y, ½ z) for the link-arc for ½ cut-off in the forward stroke. Mark the position of j r of the template at ½ j, ½ r on the diagram. In a similar manner y ½, z ½ and j ½, r ½ may be found for the position of the link-arc for ½ cut-off on the return stroke. The link is now shown in the two positions for equalized cut-off, and inasmuch as the point of suspension in locomotive practice is usually at the center of the link, the saddle-pin center must be found on the lines ½ j, ½ r, and j ½, r ½. It must, of course, be the same distance from the link-arc in each position. Therefore, locate the two points s and s_1 at equal distances from ½ r and r ½, respectively, and of such length that a line c c parallel to the central line of motion may be drawn through them. This line represents the path of the travel of the saddle-block pin,

and is in reality an arc with the hanger as the radius. For the short distance $s\ s_1$ it may be considered a straight line. Having determined the point of suspension of the link make the incision s on the template. (See Fig. 118.)

To Locate the Bell-Crank or Tumbling-Shaft for Equalized Cut-Off at All Points of Stroke.

The cut-off has now been equalized for ½ stroke where the inequality due to the connecting-rod angles is naturally at its maximum. The influence of this inequality becomes less and less as the cut-off grows later, and therefore if the maximum desired cut-off (in this case 0.92 stroke) be equalized, all intermediate cut-off positions between mid and full-gears will practically be

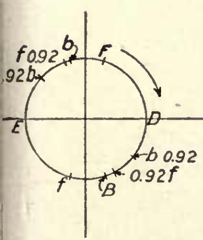


FIG. 120.

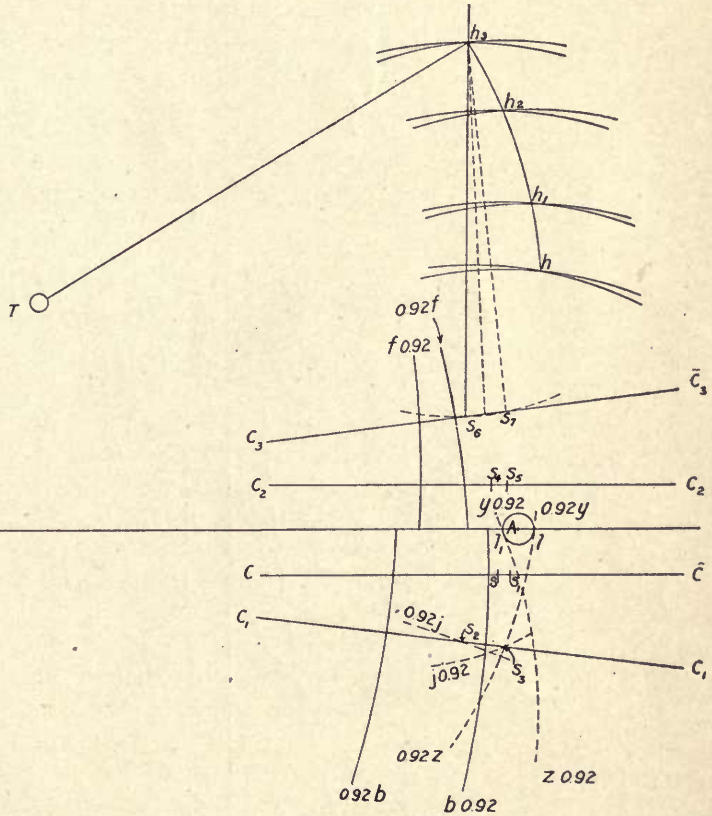


FIG. 121.

equalized. This may be accomplished by working out the proper location of the tumbling-shaft T, Fig. 121.

Figs. 116 and 117 might be used for this work, but it would complicate the diagrams too much. Therefore, on a new diagram lay off F, B, f and b with the *same values* as before; and find positions 0.92 f, 0.92 b, f 0.92 and b 0.92, Fig. 120, for the eccentric centers when the piston has traveled 0.92 of its stroke, in the *same manner* as ½ f, ½ b, etc., were found for ½ stroke. Then with the same eccentric-rod radius as before, describe the arcs 0.92 f, 0.92 b, f 0.92 and b 0.92 in Fig. 121. Adjust the template so that m and n fall on the arcs 0.92 f and 0.92 b, and the link-arc passes through l. Draw the arc 0.92 y 0.92 z and mark the point s_3 through the point s on the template. This, then,

is the position for the link at 0.92 cut-off running forward. Do the same for 0.92 cut-off on the return stroke, and mark the position s_3. Join s_2 and s_3 by a straight line c_1 c_1, which will be found to have a slight inclination to the central line of motion, but too small to produce much ill effect. It could be made parallel by placing s_2 and s_3 nearer the link-arc, but this would destroy the equality of cut-off at ½ stroke.

The saddle-pin locations s_4 s_5 and s_6 s_7 for equal cut-offs in back-gear could be found if necessary, but in most locomotive work the back-gear is a counterpart of the forward-gear, and these points may consequently be placed symmetrically with respect to s s_1 and s_2 s_3.

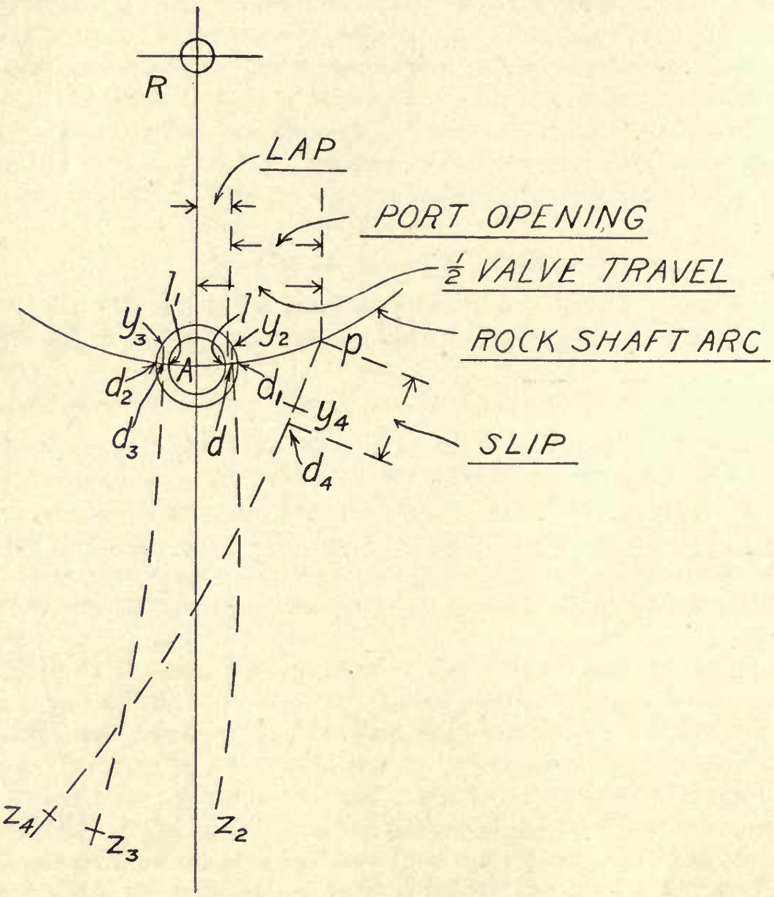


FIG. 122.

Having determined the stud positions for equalized 50% and 92% cut-offs, it only remains to suspend the hanger in such manner that for the several elevations its opposite end will sweep through the corresponding positions of s, s_1 s_2, s_3 etc. With an assumed length of hanger (which is usually determined by the space available) as a radius, and with s_2 s_3 as centers, strike arcs intersecting at h. In a similar manner, with the other three sets of points, obtain h_1, h_2 and h_3. These points will not fall on the arc of any circle, but an approximate one may be found which will give a center at T, and this point will be the center for the bell-crank shaft.

To Find the Lead on the Forward and Return Strokes in Full-Gear.

With the points F, B, f and b sweep arcs F, B, f and b on a new diagram (not shown here) similar to those in Fig. 117, and adjust the template with m and n on F and B, and with s on c_1 c_1 of Fig. 121. Mark the point in which the link-arc intersects the line of motion. (Figs. 120 and 121 are necessarily drawn on such a small scale that instead of further complicating these figures by drawing in this con-

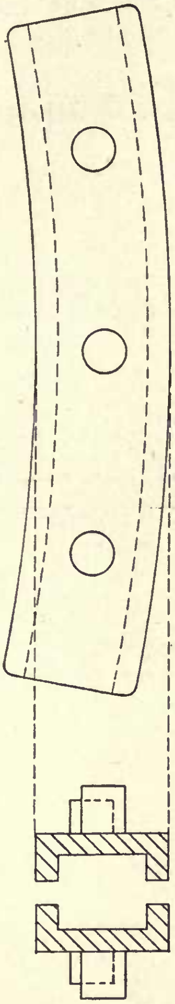


Fig. 123.

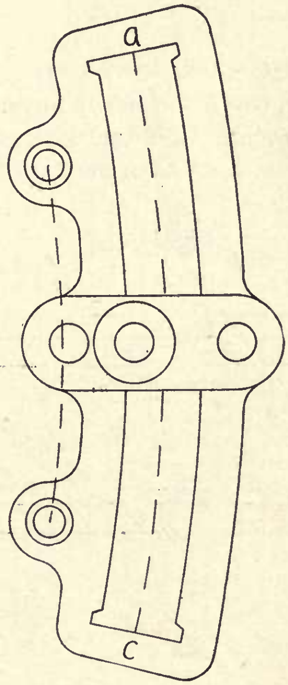


Fig. 124.

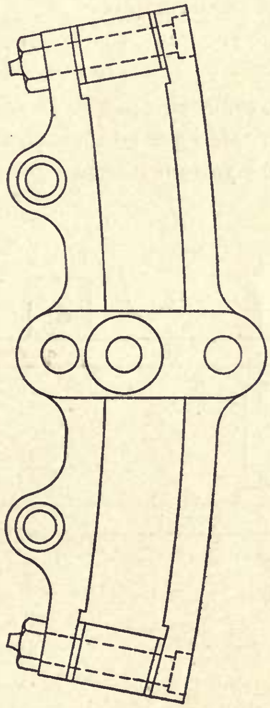


Fig. 125.

struction we will show the *results* of the construction called for in this paragraph in the separate Fig. 122.) The distance of this point to A, minus $A\ l$, will be the lead (equal $l\ d$) at full-gear on the forward stroke. In the same manner, by using arcs f and b, the lead on the return stroke full-gear may be obtained equal to $l_1\ d_3$. These leads ($l\ d$ and $l_1\ d_3$) will be found to be slightly unequal, but on account of the large port-opening at full-gear, the effect of their inequality may be neglected.

To Find Extreme Travel of the Link and the Slip.

With the template in position on the arcs f and b as called for in the previous paragraph, the point y' (Fig. 118) on the link-arc will have its greatest elevation for full-running position, as represented at

$y_3\ z_3$ in Fig. 122. The forward eccentric has its greatest throw when F is at D, Fig. 120, then B is to the left an amount corresponding to the arc $F\ D$, and the link is in its greatest inclined position, as represented by the link-arc $y_4\ z_4$ in Fig. 122. For this position the point p of the link-arc is on the rock-shaft arc, and is the distance $p\ y_4$ from y' on the template. For position $y_3\ z_3$ the point d_3 of the link-arc is on the rock-shaft arc. The maximum slip is therefore $p\ d_4$.

For further reference in laying out this link-motion graphically, see "Link and Valve Motions," by Auchincloss, pages 90 to 135.

Use of Models in Construction of Valve-Gears.

Models are sometimes built, and the required valve motion obtained by adjustments in the several parts of the model.

Links.

Classifications and Types.

Links, in general, may be classified in two independent ways:

1. With reference to their suspension, into "shifting" and "stationary" links.
2. With reference to their form, in which we have the "box" link, Fig. 123; the "open" link,

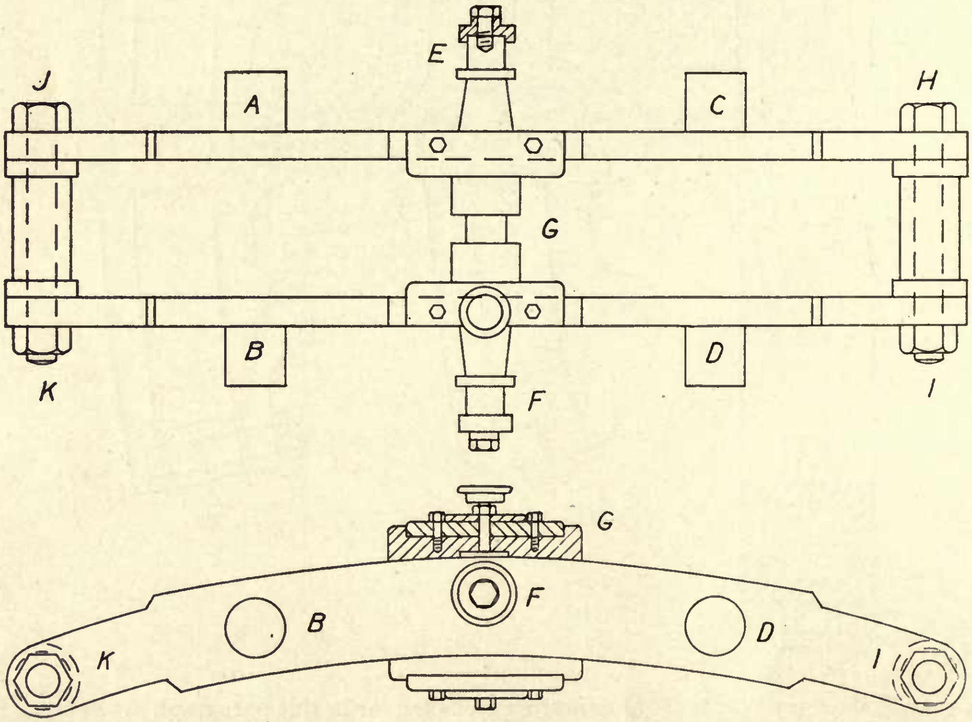


Fig. 126.—Stephenson Double-Bar Link.

solid, Fig. 124; the "open" link, built up, and more generally known as the "skeleton" link, Fig. 125, and the "double-bar" link, Fig. 126.

Shifting and Stationary Links.

The "shifting" link is represented in the Stephenson gear, Fig. 108; the "stationary" link in the Gooch gear, Fig. 129. A "shifting" link is distinguished by the fact that the link itself is moved up or

down to secure variable cut-off or reversal; in the "stationary" link the "radius-rod" instead of the link is moved to secure variable cut-off or reversal. Both the shifting and stationary links have similar motions throughout a cycle.

Forms of Links in General Use.

The forms of links most used in American practice are shown in Figs. 124, 125 and 126. In the link shown in Fig. 124 the eccentric-rod pins may be placed on extensions of the link-arc $a\ c$, in which

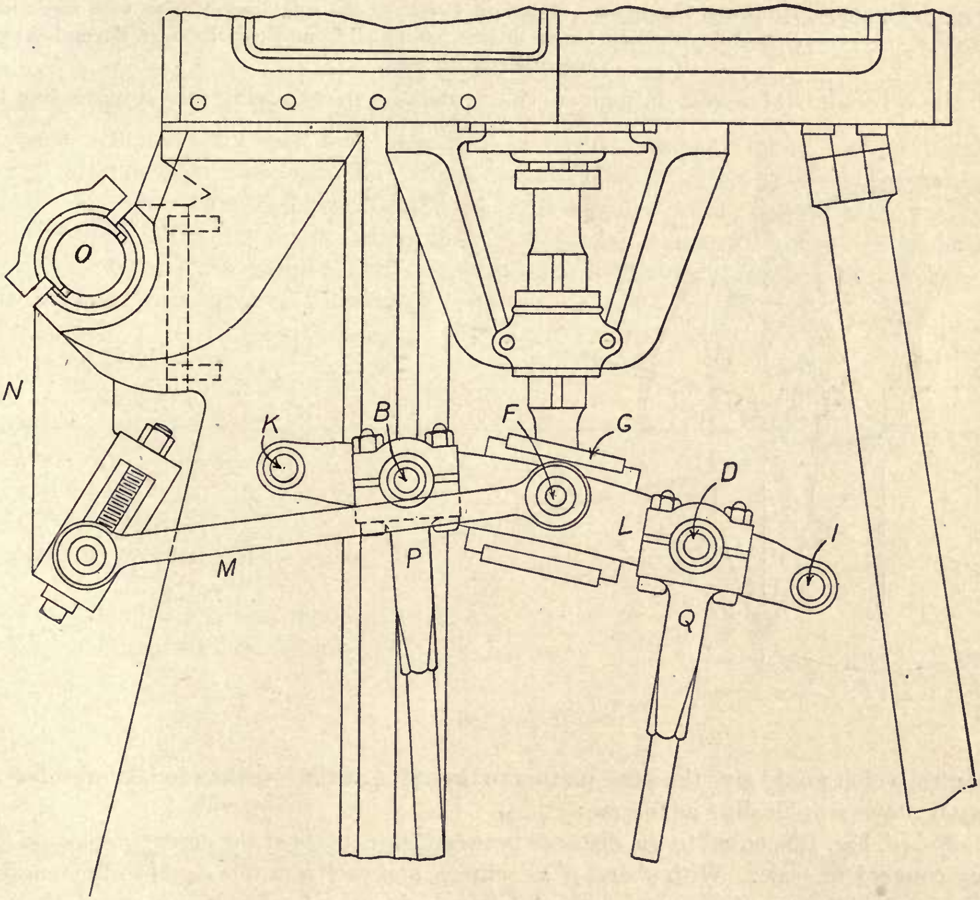


FIG. 127.—Arrangement of Stephenson Link and Rock-Shaft Connections.

case the diameter of the eccentric circle must be greater than the travel of the valve. The "double-bar" link, as shown in Fig. 126, is applied principally to marine engines; its action is shown in Fig. 127, in which L is the link, P and Q the eccentric-rods, M side or bridle-rods, N the rock-shaft arm, O the rock-shaft, or weigh-shaft.

The end of the rock-shaft arm is provided with a block operated by a screw so that the valve may be adjusted without moving the weigh-shaft. The line of motion of this block is so designed that when the link is in position for full-gear forward the movement of the block will be in line with the bridle-bars, and any adjusting motion communicated without loss.

The eccentric-rod P (Fig. 127), by means of forked ends, is connected to the pins A and B, Fig. 126, and similarly Q is connected at C and D. G is the link-block through which the link slides, and to which the valve-stem is directly attached; and E and F, the pins to which the bridle-rods are connected. The pins E and F are independent of the link-block, and may be placed at the center as shown, or at the ends as extensions of $H\ I$, or $J\ K$, or at intermediate positions according to the requirement of the design.

Drafting Table Problem, No. 7.—Comparison of Results From Open and Crossed-Rods.

Comparison of Theoretical Indicator-Cards for a Standard Locomotive Using Link-Motion with Open-Rods, and Cutting off at.......... Stroke, with those for an Engine having all Conditions the same Excepting that Crossed-Rods are used.

With the valve and valve-gear data given, this problem is most readily solved by finding the vir-

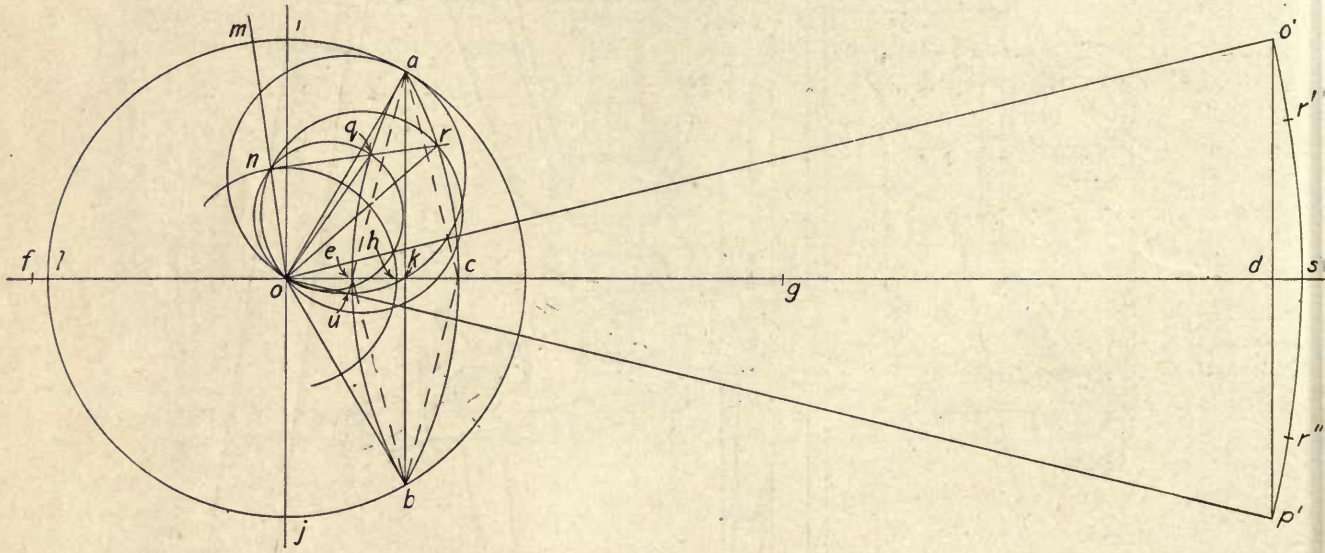


Fig. 128.

tual eccentric which would give the same motion to the valve as the link does for the specified cut-off. This may be done graphically [1] as follows:

Lay off $o'\ p'$, Fig. 128, equal to the distance between the centers of the eccentric-pins on the link, using any convenient scale. With o' and p' as centers, and with a radius equal to the length of the eccentric-rod, describe arcs intersecting at o and draw $o\ o'$ and $o\ p'$. Through o and d (the center of $o'\ p'$) draw $f\ o\ d$, the central line of motion of the valve-gear.

With a radius $o\ a$ equal to the radius of the eccentric, draw the eccentric circle $a\ b\ l$ to any convenient scale. Describe the lap circle $o\ h$; lay off the full-gear lead $h\ k$; and draw $a\ k\ b$ perpendicular to $o\ d$, thus obtaining a and b, the positions of the eccentric-centers for full-gear.

$a\ o\ i$ = the angle of advance. $o\ a$ and $o\ b$ are the diameters for Zeuner circles for the link in full-gear, for either open or crossed-rods. With open-rods $o\ a$ will be the position of the eccentric radius for going forward (*i.e*, running "under") and $o\ b$ for backing (*i.e.* running "over"), it being kept in mind that a reversing rocker is used and that the crank is at $o\ c$.

[1] For complete graphical demonstration, see "Designing Valve-Gearing," by E. J. C. Welch, pp. 105 to 141.

The travel of the valve, and the events of the stroke are thus determined by the Zeuner circle $a\ k\ o$, for full-gear. With the slide-block at any other position, such as at r', the virtual eccentric may be found as follows for forward running:

Draw $a\ c$ perpendicular to $o\ o'$; this perpendicular, extended, cuts the central line of motion at c and gives $h\ c$ as the mid-gear lead. A circular arc drawn through the points $a\ c\ b$ corresponds closely with the curve containing the loci of the extremities of the diameters of the Zeuner circles (or the centers of the virtual eccentrics) for all intermediate positions. Therefore the virtual eccentric for the slide-block at r' has a radius $o\ r$, found by making $a\ r : a\ c\ b :: o'\ r' : o'\ s\ p'$, and the Zeuner circle $n\ r\ o$ determines all the events of the stroke but not crank positions. They will only be determined if the Zeuner circles are in the proper quadrant. If the position of the slide-block is desired for a given cut-off as, for example, with the crank at $o\ m$, r is found by drawing $n\ r$ perpendicular to $o\ m$ and tangent to the lap-circle; and r' by the proportion just given.

If the link is actuated by crossed-rods the slide-block (represented at r' for open-rods) would be at r'' ($r''\ p' = r'\ o'$) to give the same cut-off as before, and the Zeuner circle $n\ q\ o$ would show the steam distribution. The arc $a\ e\ b$ is found by drawing $a\ u$ perpendicular to the mean eccentric position $o\ p'$ for crossed-rods, and noting the point e where the perpendicular crosses the central line of motion.

The necessary data for this problem may be taken from the first eight items of the following table of dimensions of the valve-gear of a locomotive:

Stroke of piston	24″
Maximum travel of valve	5½″
Steam-lap	1¼″
Exhaust-lap	0
Lead, full-gear	1/16″
Length of connecting-rod	92″
Length of eccentric-rods	57½″
Distance apart of eccentric-pins	12″
Distance of eccentric-pins behind link-arc	3″
Distance of tumbling-shaft from main shaft	44″
Radius of tumbler	17″
Radius of hanger	13½″
Tumbling-shaft above main shaft	12½″
Height of rock-shaft above main shaft	8¾″

Mid-gear lead same on both strokes.

Enter a table of results on the plate as follows:

	Per Cent. of Completed Stroke				Lead.
	Admission	Cut-off.	Release.	Compression	
... open-rods					
... crossed-rods					
... open-rods					
... crossed-rods					

NOTE: Take initial boiler pressure of 85 lbs. gauge with 2 lbs. back pressure and a 40 spring. Assume a clearance volume of 5%. Make the indicator cards 4 inches long.

Types of Valve-Gears.

Gooch Gear.

The link for this valve-gear is a "stationary" one and is shown in Fig. 129. The characteristics of the gear are:

1st. That with the engine on dead-center the slide-block moves up and down, while the link remains stationary.

2d. That the link curvature is *convex* to the engine-shaft, and has for the radius the length of the radius-rod.

From the method of construction of this form of gear it will be observed that the slide-block may be moved from one end of the link to the other without altering the position of the valve. This means that the *lead* opening (shown in the sketch of the valve section, Fig. 129), is constant for all positions of the slide-block in the link. With the Stephenson link the lead opening is depend-

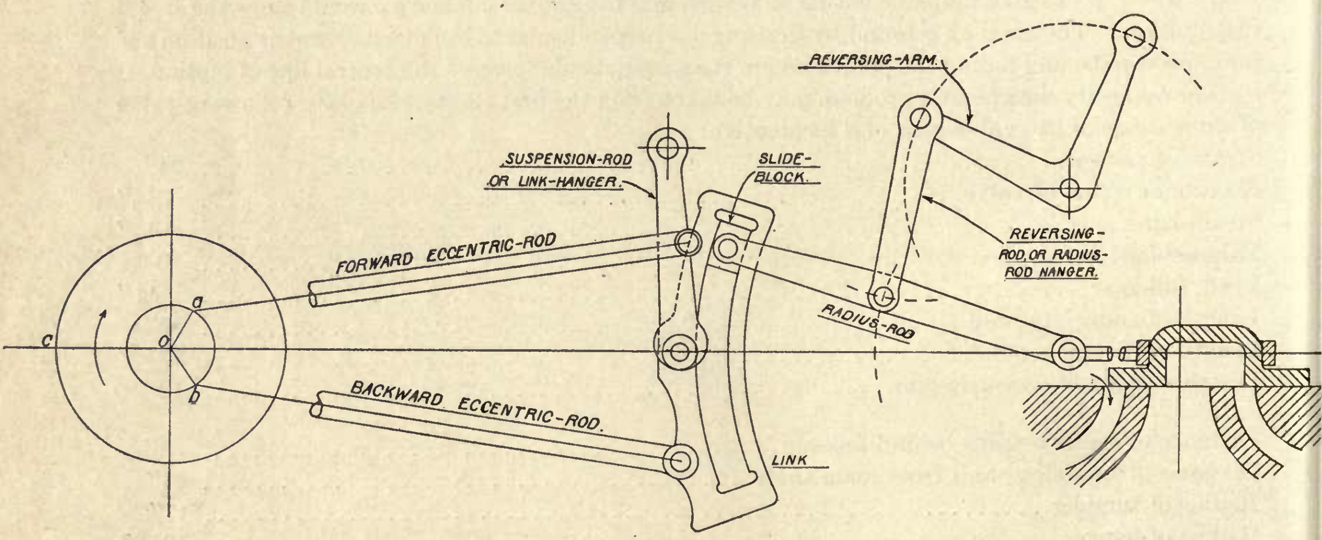


Fig. 129.—Gooch Gear.

ent on the arrangement of the eccentric-rods, but with the Gooch link the result remains the same whether open or crossed-rods are used.

The Gooch gear takes much more space than the Stephenson gear on account of the radius-rod.

For stationary engines the Gooch gear is especially adapted for use in connection with a governor, for the reason that the radius-rod throws a much less, and more easily balanced load on the governor than does the shifting link with its rods, hanger, additional friction, etc.

Allen Gear.

The special features of this form of gear are the straight-line link $k\ n$, and the simultaneous operation of the link and the radius-rod ($k\ l$) through the suspension-rods $f\ g$ and $d\ h$ pivoted to the rocker-arms $e\ f$ and $e\ d$, Fig. 130.

The main object in laying out a design using the Allen link is to so proportion the lengths of these reversing-arms that, as the link moves up and the radius-rod down, the point k will move

to k_1 in as nearly an arc about l as possible. If it moved in a circular arc the valve would have a constant-lead opening, as in the Gooch motion. The Allen link gives less variable lead than the Stephenson, and with long eccentrics and radius-rod the lead is practically constant. Properly

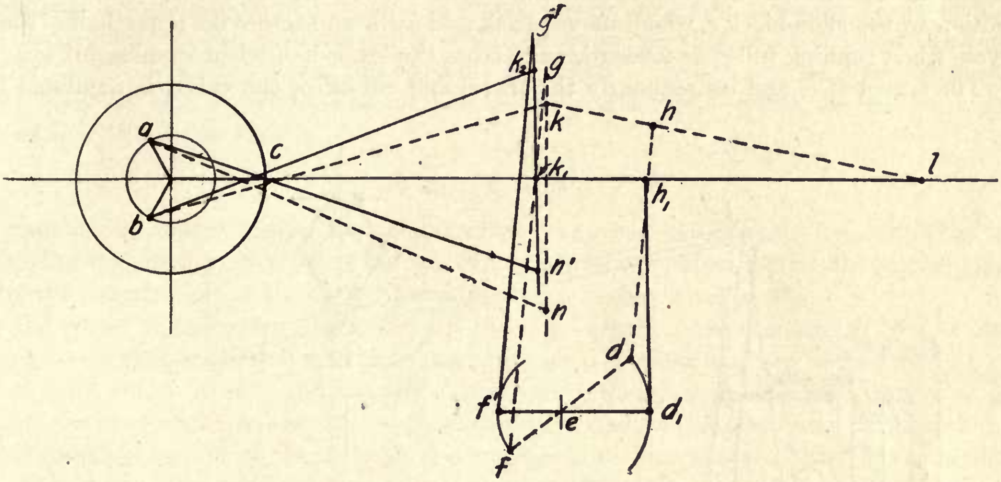


Fig. 130.—Allen Gear.

designed reversing-arms tend, incidentally, to equalize the moments on the two sides of the reversing-shaft e. The sketch is somewhat distorted to avoid overlapping of lines; $b\ k_2$ should equal

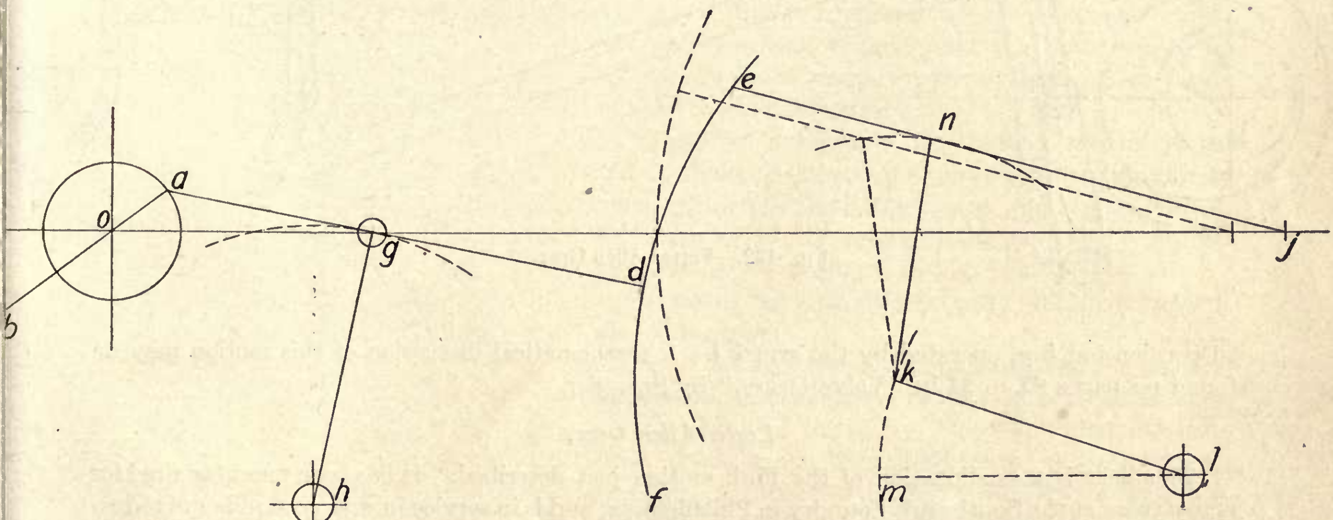


Fig. 131.—Fink Gear.

$b\ k$, and $a\ n'$ should equal $a\ n$. Directions for proportioning these arms may be found in "Link and Valve Motions," by Auchincloss, pages 140 to 142.

Fink Gear.

Fig. 131 shows a center-line diagram of the Fink gear. The point o is the engine-shaft, $o\ a$ the eccentric-arm, and $o\ b$ the crank in line with $o\ a$. The eccentric-rod $a\ d$ is rigidly connected at

d to the link-arc *e f*. The radius-rod *e j* connects at *j* with the valve-stem. The arc *e f* is drawn with *e j* as a radius so as to maintain a constant lead at all gears. The point *g* in the eccentric-rod is designed to move in an arc coinciding as nearly as possible in the line *o j* by being pivoted to the radial arm *g h* as shown. This mechanism causes the link-arc to move up and down and thus give motion to the slide-block *e*, which moves back and forth and across the upper half of the arc of each cycle when running full-gear forward, and across the lower half when running full-gear backward. The travel of *e*, and consequently the travel and cut-off of the valve, is regulated by the

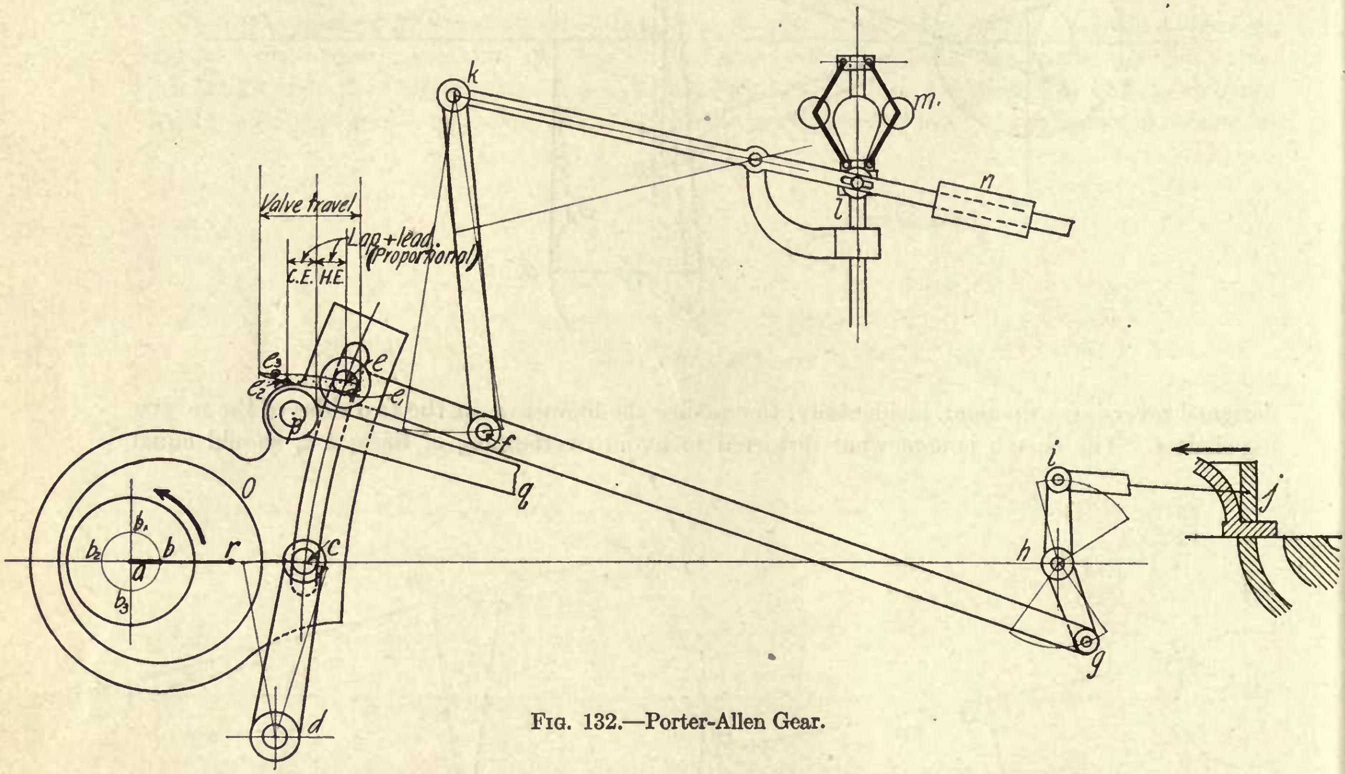


Fig. 132.—Porter-Allen Gear.

suspension-rod *k n*, operated by the arm *k l*. A mathematical discussion of this motion may be found on pages 87 to 94 in "Valve Gears," by Spangler.

Porter-Allen Gear.

This gear is a modification of the Fink motion just described. It has been manufactured for many years at the Southwark Foundry in Philadelphia, and is in service in a large number of industrial plants throughout the country.

The eccentric is represented by the heavy weight line *a b* in Fig. 132, and the crank by the medium weight line *a r*, and both are set in the same direction. The center of the eccentric-sheave is at *b* and the circle b, b_1, b_2, b_3, is the path of the eccentric-center. If, in the Fink gear, the point *d* is moved back to coincide with *g*, the principle feature of the Porter-Allen motion is obtained. The latter gear is usually made to give variable cut-off only and therefore the reversing arc (represented by *d f* in the Fink motion) is omitted in Fig. 132. The Porter-Allen gear operates separate

live steam and exhaust valves, the latter through the rod $p\ q$, which it will be observed is not adjustable and therefore gives constant release and compression for all cut-offs.

The path of the point e of the *eccentric-strap arm* is represented by the closed curve, e, e_1, e_2, e_3, while the path of the point e of the *rod* $e\ g$ is represented by a curved line not shown in the sketch, but which the student should be prepared to draw. Since these two paths do not coincide, there will be continuous slipping, or a quivering action, between the slide-block pin and the guide-arc at e, such as is common to curved link-motions generally. The arc $e\ c$ has g for its center and therefore constant lead is obtained at all cut-offs. The manner in which the governor controls the cut-off is shown in the sketch.

Walschaert Gear.

The mechanism composing the Walschaert valve-gear is entirely different from any thus far considered. The resultant motion of the valve is due to two independent component motions, one produced by the eccentric-pin, c, the other by the crosshead, as shown in Fig. 133.

a is the center of the engine-shaft, and $a\ b$ the main crank. The eccentricity, $a\ c$, is obtained by keying the eccentric crank $b\ c$ to the main crank-pin, b, outside of the connecting-rod. $a\ c$ is taken at right angles to $a\ b$, and the angle of advance, therefore, is zero: this means, of course, that so far as the eccentric motion is concerned the valve could have neither lap nor lead and steam would be admitted for full stroke, as explained on page 2 of these notes. The link $r\ s$ oscillates on a fixed shaft shown at k in Fig. 133 and at m_1 in Fig. 134. Any desired valve-travel and cut-off for either forward or backward motion of the valve may be obtained by shifting the slide-block k (attached to the radius-rod) along the link $r\ s$, by means of the radius-rod hanger.

The arm $d\ e$, which is firmly fixed to the crosshead at one end, connects at the other by means of a connecting link with the lap and lead lever $f\ g\ h$. This lever so combines the component eccentric and crosshead motions that the latter makes up for the angular advance which was neglected in laying out the eccentric-center c.

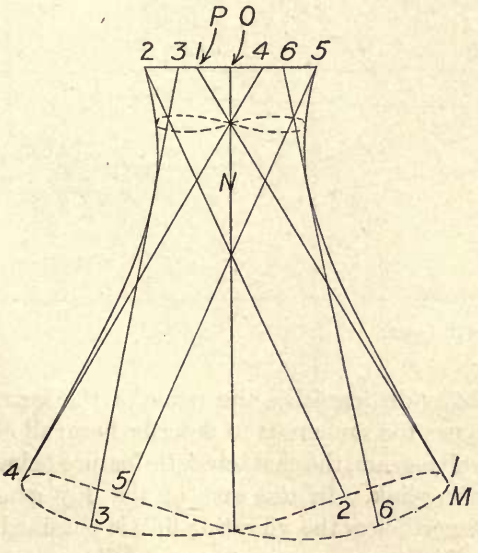


FIG. 135.

A general analysis of this motion may be carried out by dividing the eccentric and crank circles into an equal number of parts, starting at c and b and finding, by construction, the corresponding positions of the lap and lead lever, as shown in Fig. 135.

In laying out and adjusting the Walschaert gear it should be noted:

1. That in order to get constant lead for all running positions, the link-arc $r\ s$ must have a radius equal to the length of the radius-rod $g\ k$ and that when the main crank is on either dead-center the connections through the eccentric-crank, eccentric-rod and link must be such that the link-arc $r\ s$ has the corresponding position g of Fig. 133 as a center. Then, no matter where the link-block k may be located, whether at the extremes for full-gear (k_1 Fig. 134), or the center for mid-gear (k, Fig. 133), the lead will be the same, for k may be moved along the link, when in the position just described, without moving the valve.

2. The lap and lead lever should be vertical when the piston is at the middle of the stroke and

the radius-rod in the mid-gear position; also its length should be chosen so that its angular vibration shall not exceed 60 degrees, preferably 45 to 50°.

3. The rod *e f* should vibrate through equal angles above and below a horizontal line.

Radial Valve-Gears.

With the Walschaert gear, just described, and the Hackworth and Marshall gears about to be taken up, it will be noticed that variable travel of the valve, with consequent variable cut-off, and also forward and backward running, are obtained with the use of only one eccentric or its equivalent. The final motion given to the valve-stem in each case is the resultant motion of that due to the

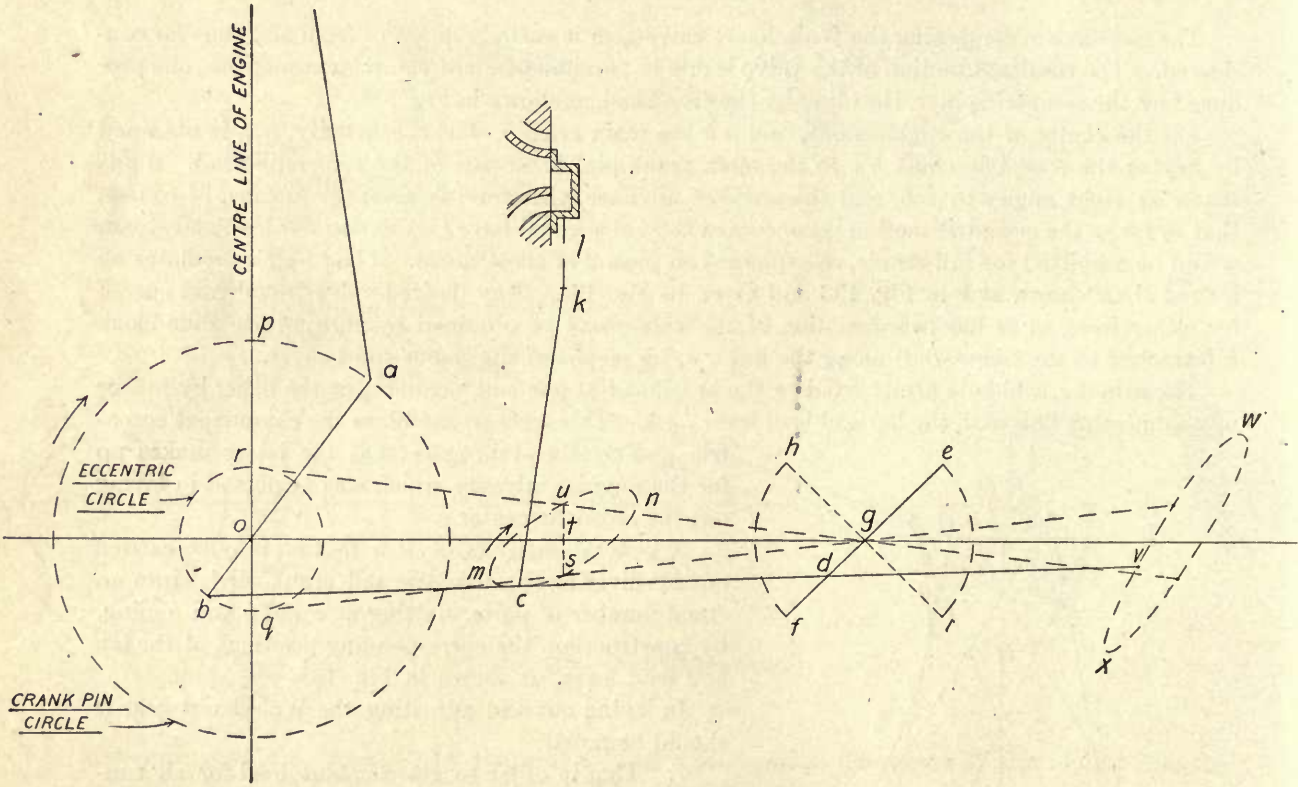


Fig. 136.—Hackworth Gear.

eccentric, and to some other mechanical feature, which latter distinguishes the name of the gear. In addition to the gears just mentioned there are other types too numerous to describe here; all of this style are frequently grouped under the head of radial valve-gears, the characteristic feature being that the resultant motion of the valve is taken from a vibrating-link. In the case of the Joy gear soon to be described there is not even one eccentric, but nevertheless the vibrating-link is obtained.

The general advantages of radial valve-gears are: Lightness, compactness, small number of moving parts, and constant lead. The general disadvantages are: Unequal valve motion, unless vibrating-lever is long (Hackworth gear excepted), large transverse stress on vibrating-link in case of an unbalanced valve, or of high speed.

Hackworth Gear.

In Fig. 136 $o\ a$ is the engine-crank, and $o\ b$ the eccentric which, in this gear, is always set either with the crank, or 180° from it. $b\ d$ is the vibrating-link and is of constant length; $e\ f$ a slide-bar pivoted at the point g; $k\ l$ the valve-stem and $k\ c$ the valve-stem connecting-rod. $c\ m\ n$ is the path

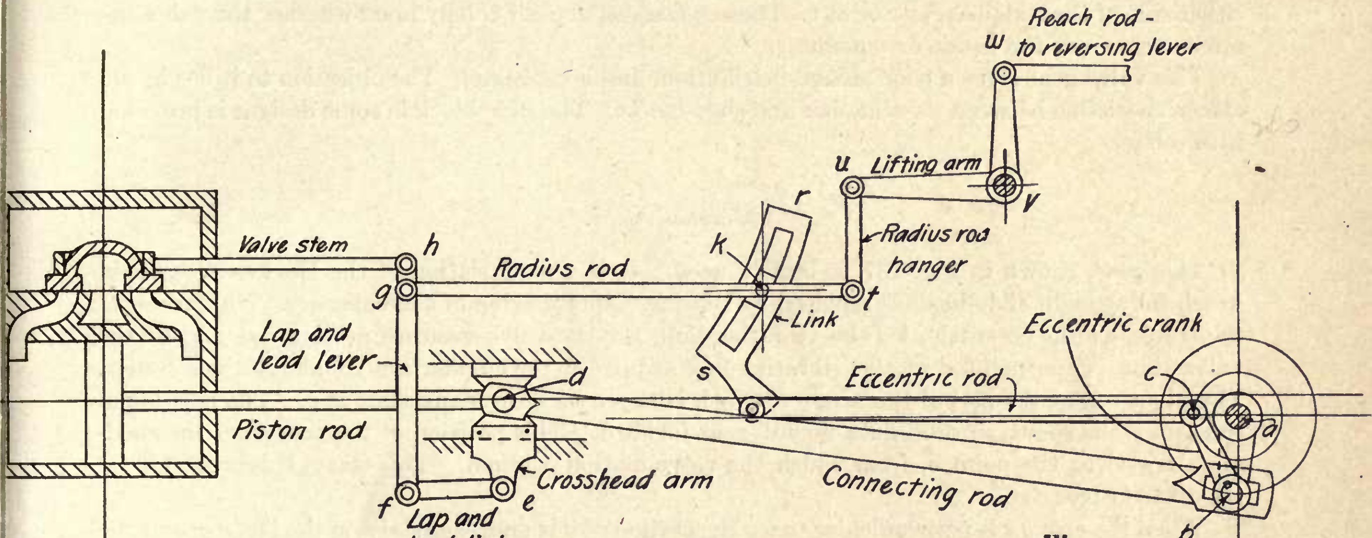


FIG. 133.

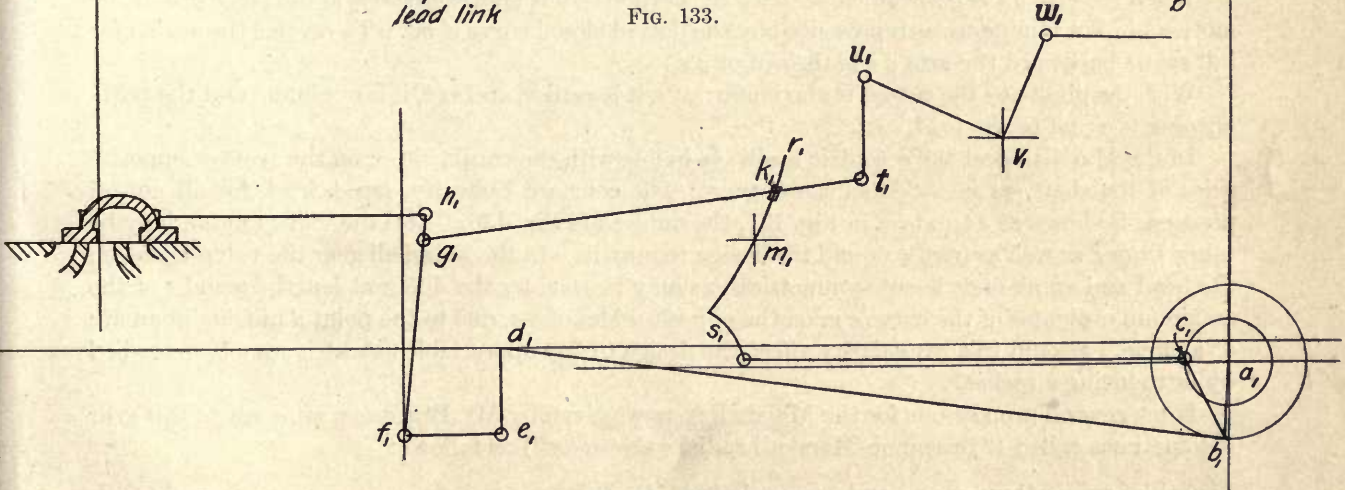


FIG. 134.—Walschaert Gear.

of the point c. The fixed point, d, on the vibrating-link travels forward and back on the slide-bar once during each revolution.

By adjusting the inclination of the slide-bar, the resultant vertical motion of the valve is modified, and the point of cut-off varied. When the slide-bar is horizontal the valve motion is a minimum;

when its inclination is reversed, as shown at *h i*, the engine is reversed. *o p* is the outward dead-center position of the crank. When the crank is on dead-center the vibrating-link is at *q g*, or *r g*, and the valve is off center a distance *s t* or *t u*. These distances are the same, and are equal to the lap plus the lead. Therefore if the lap is the same on both ends of the valve, the lead is the same and is constant for all running positions, and is independent of the inclination of *f e*. According to the requirements of the design the eccentric, which must be in line with the crank, may be either on the opposite side or on the same side; and the valve motion may be taken from the vibrating-link on either side of the slide-bar, at *c* or at *v*. These selections depend chiefly upon whether the valve admits steam from the inside or outside.

This valve-gear gives a good steam distribution, and is compact. The objection to it lies in the excessive friction between the slide-bar and slide-block. The slide-block in some designs is provided with rollers.

Marshall Gear.

This gear, shown in Fig. 137, is largely used. It is a modification of the Hackworth gear in which the straight slide-block is replaced by a swinging pin moving in a circular arc. *o a* represents the crank, *o b* the eccentric, *b d* the vibrating-link, *k c* the valve-stem connecting-rod, and *l k* the valve-stem. The point *d* of the vibrating-link swings in the circular arc *f h* about *e* as a center. The pivot *e* is at the end of the arm *g e*, which is keyed to a reversing shaft at *g*. The position of the arm *g e* is shown in solid lines for full-gear forward. This position of the arm gives the maximum travel to the point *c*, from which the valve motion is taken. This travel is represented by the dotted curve *c m*.

When the arm *g e* is perpendicular to *o g* the motion of *d* is approximately on the line *o g*, and the motion of *c* is a minimum, as represented by the dotted closed curve c' m'. To reverse the engine for full speed backward the arm *g e* is thrown to $g\ e_4$.

With the pivot at *e* the cut-off is maximum; at e_1 it is earlier, and at e_2 it is minimum and the port-opening is equal to the lead.

In the Marshall gear the eccentric is always in line with the crank, either on the same or opposite sides of the shaft, as in the Hackworth gear. The constant quantity, lap + lead, for all cut-off positions is shown at *s t* and *t u* in Fig. 137, the same as in Fig. 136. Also the valve motion may be taken from *x* as well as from *c* should the design require it. In the Marshall gear the valve-travel on the head and crank ends is not symmetrical, as may be seen by the different lengths *y* and *z* of the maximum ordinates of the curve *c m* on the opposite sides of *o g*, due to the point *d* moving in an arc of a circle. Should this irregularity affect the design to any appreciable extent it may be remedied by introducing a rocker.

Some general proportions for the Marshall gear are given by Mr. Braemme, after whom this gear is sometimes called ("Braemme-Marshall radial valve-gear") as follows:

Length of supporting arm *g e* and suspension-rod *d e*.	$= 6 \times o\,b$.
Eccentric-rod *b d* (exaggerated in Fig. 137)	$= 6 \times o\,b$.
Lead arm *d c*.	$= 4.5 \times o\,b$.
Angle α should not be more than 25°	

In connection with the Hackworth and Marshall gears the student will be required to assume the data given in the first two columns of accompanying table and to fill out columns 3 and 4 and draw a center-line sketch of either gear to illustrate the work.

1 Angle between crank and eccentric.	2 Location of c	3 Kind of valve admission. Inside or outside.	4 Direction of rotation of engine with d moving from upper left to lower right.
0°	Left of d		
0°	Right of d		
180°	Left of d		
180°	Right of d		

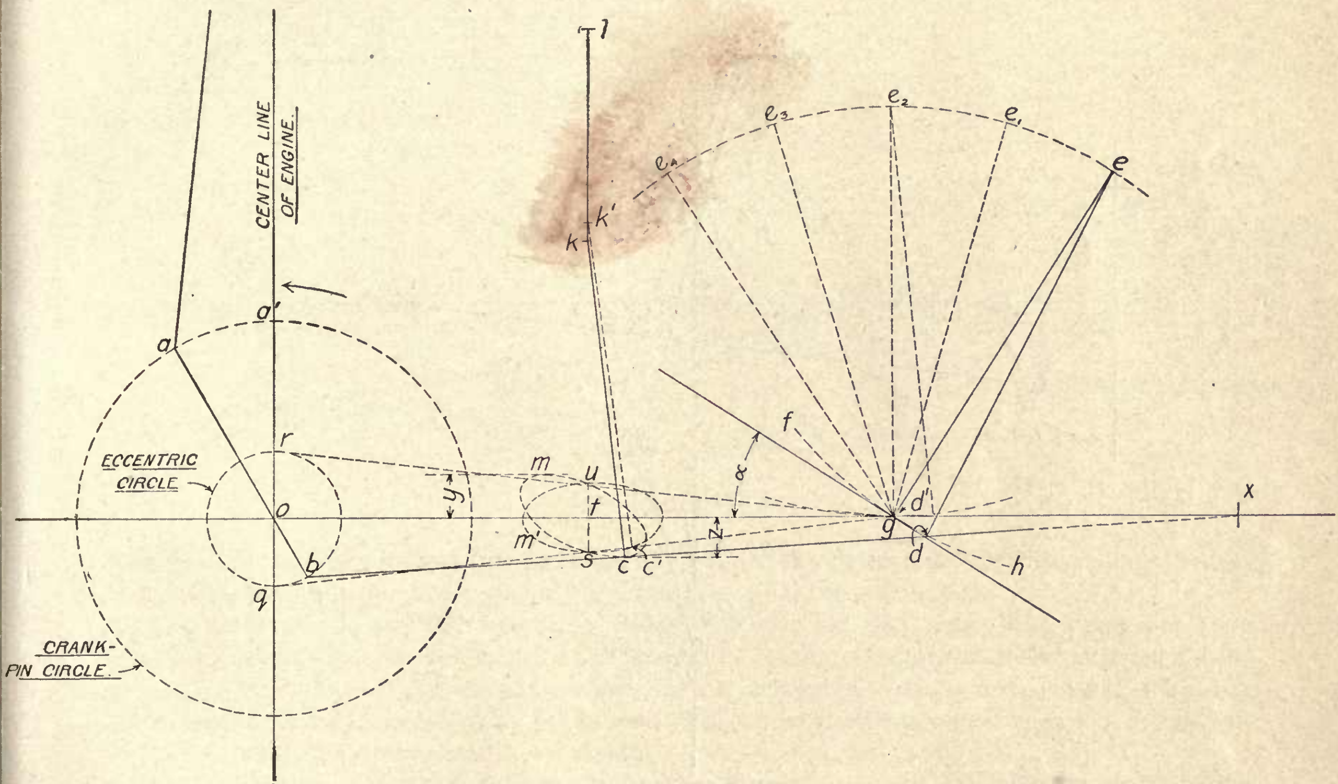


FIG. 137.—Marshall Gear.

Joy Valve-Gear.

This gear, sometimes called a "compound radial gear," does away with the eccentric altogether, the valve motion being obtained solely from the connecting-rod by a series of rods or arms.

o a, Fig. 138, represents the crank, *a b* the connecting-rod, *c e* and *d k* vibrating-rods, *e f* an arm of which the point *e* moves always in arc about *f* as a center, *i j* (a guide-arc for *h*) is pivoted at *g*, and constructed so that it may be temporarily fixed in any position, as, for example, that shown by the dotted position, i_1 j_1. The position of this guide-arc determines the point of cut-off. The dotted ovals through *c* and *d* show respectively the paths of these points, no matter what the cut-off

gear. The oval through k shows the path of k for the position $i\ j$ of the guide-arc. This oval varies for different cut-offs, and in the mid-gear position the tangent to $i\ j$ at g would be a vertical line, thus giving a minimum travel to k and to the valve, which should equal lap plus lead.

When the engine is on dead-center d is at d_1, h at g, and k at k , and the horizontal distance from k_1 to the vertical center-line shows the amount the valve is off center, which in the dead-center position equals lap plus lead. The engine is reversed with this gear by swinging the guide-

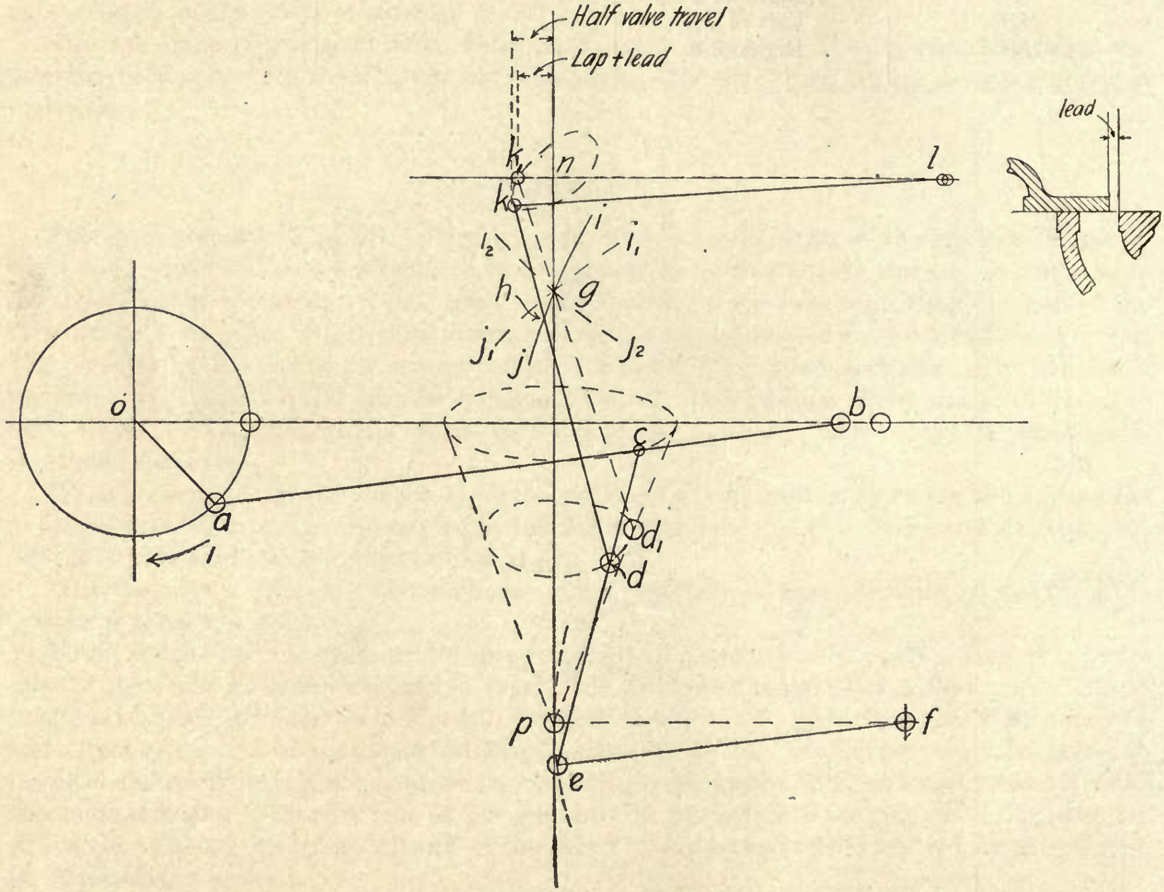


FIG. 138.—Joy Gear.

arc about g beyond the mid-gear position to $i_2\ j_2$. In the dead-center position the line through p and g should be perpendicular to $o\ b$.

The effect of the angularity of the rod $k\ c$ in the Marshall gear (Fig. 137) is partially neutralized with the Joy gear by the vibrating-rod $e\ c$. When properly proportioned the Joy gear gives a rapid motion to the valve when closing the ports, less compression at short cut-off than a Stephenson link motion, and a nearly equalized cut-off for all grades of the gear. It gives a constant lead. These points, favorable to the Joy gear, are counterbalanced in part by the number of parts and joints that are liable to give trouble with wear, and the obstruction it offers to proper care and attention.

It will be noticed that the Joy valve-gear is practically the same in construction and principle as the Hackworth and Marshall gears from the point d to l. The path of d in the Joy gear takes the place of the eccentric in the other two.

Baker Gear.

This gear has been very recently developed in connection with American locomotive construction. It not only does away with the eccentric but also with the curved link, and there is no sliding friction whatever in the gear. Illustrated in Fig. 139, it will be seen that the part of the mechanism from a to j contains the crosshead and return-crank drive characteristics of the Walschaert gear, while the remaining parts are suggestive of the Marshall gear.

The Baker gear is a modification of the Baker-Pilliod gear which came into use in 1908, but its

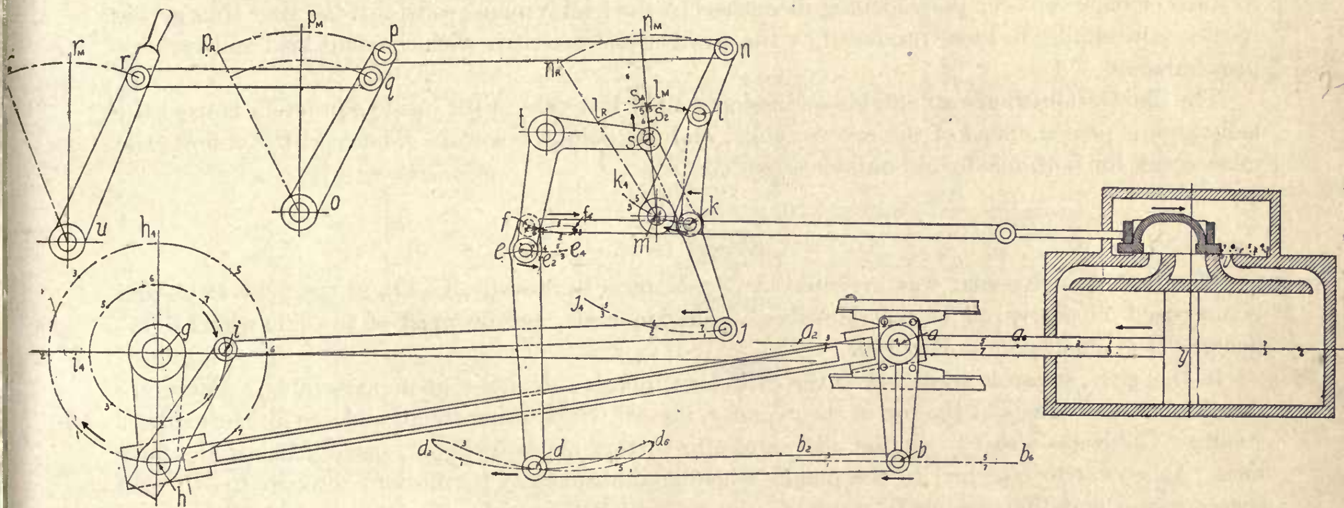


FIG. 139.—Baker Gear.

manufacture was discontinued two years later in favor of the Baker gear, notwithstanding the fact that forty-three railroads had installed the original gear during that period.

The names of the gear parts are: Crosshead arm, $a\ b$, Fig. 139; union link, $b\ d$; combination lever $d\ e\ f$ (all one piece); bell-crank, $e\ t\ s$ (the arm $e\ f$ on the combination lever falls behind the arm $e\ t$ of the bell-crank for the phase shown in the illustration); gear-connection rod, $s\ k\ j$ (one piece); radius-bar, $k\ l$; reverse yoke, $m\ l\ n$; reach-rod, $p\ n$; reverse arm $o\ p\ q$; reach-rod, $q\ r$; reverse lever, $r\ u$; crank, $g\ h$; return-crank, $h\ i$; eccentric-rod, $i\ j$; connecting-rod, $h\ a$.

The mechanism is shown in position for full-gear forward. In order to follow more closely the motion of the various parts during one cycle, the cycle has been marked at six phases and the paths and directions of the several points drawn. All fixed centers are indicated by vertical and horizontal center lines.

When running forward the reverse yoke $m\ n$ remains stationary. To give earlier cut-off $m\ n$ is thrown over toward $m\ n_M$; and to run backward it is thrown beyond n_M until full-gear backward is reached at $m\ n_R$.

The pivot, i, it will be noted, is fixed 90° behind the crank and therefore has zero angle of advance, and the motion from it alone would call for an elementary valve without lap, and would admit steam for full stroke. The motion from the crosshead gives the additional travel to the

valve necessary to make up for lap and lead, and in this general respect the Baker and Walschaert gears are the same, although the actual valve motions at the succeeding phases of the travel are different in the two gears, thus permitting different claims to be made for the respective gears. These claims may be followed and analyzed from actual measurements of the gears, or from working drawings, by following the paths of the various points in a manner similar to that shown in Fig. 139 when drawn to a greatly enlarged scale.

When the mechanism is set at mid-gear, with $m\ n$ at $m\ n_M$, the arc of swing, $k\ k_4$, will have l_M for its center and s will remain stationary at s_2. With s_2 stationary, e will also remain at rest at e_2 and the eccentric will impart no motion to the valve. Under these conditions the only motion the valve has comes from the crosshead, the gear being so proportioned that the half valve-travel is then equal to lap plus lead. As the reverse gear is thrown from the mid-gear position either forward or backward the port opening increases but the lead remains constant, the gear thus giving results quite similar to those produced by the straight-slot eccentric with constant lead and variable preadmission.

The illustration shows an outside admission gear. If a valve with inside admission is used the bell-crank is placed ahead of the reverse yoke, and the point f below e. The eccentric follows the main crank for both inside and outside admission.

Stevens Gear.

The Stevens valve-gear was invented by Mr. Francis B. Stevens, E. D., in the year 1839, and is now used on nearly all of the side-wheel excursion craft, and on most of the side-wheel ferryboats. It is illustrated in Fig. 140.

In this gear, steam is admitted to the cylinder through a double-seat poppet-valve. There are two double-seat valves at the top of the cylinder, one for the entering steam and one for the exhaust steam. There are also two similar valves at the bottom of the cylinder, usually below the floor line. An eccentric attached to the paddle-wheel shaft transmits its motion, through the trussed eccentric-rod and the rock-shaft crank, to the rock-shaft to which are rigidly attached cams, or wipers, as they are usually called. These wipers work against toes, which are rigidly attached to the steam and exhaust-rods. These, through the valve-lifter, raise and lower the double-seat valves.

On the large excursion steamers one eccentric only is generally used for the live steam and one for the exhaust. On ferryboats there are two live-steam eccentrics, one for going forward and one for going backward, and also two eccentrics operating the exhaust. Where only one eccentrcis used for live steam the valve must be operated by hand while the engine is backing.

In order to start an engine having this gear, it is necessary for the engineer to operate the valve through his own effort. This is accomplished through the starting-bar lever and the auxiliary, or starting rock-shaft, to which are attached a duplicate set of wipers, in miniature, operating on auxiliary toes on the steam-rods. The effort required for this work is not excessive, as the double-seat valve is practically a balanced valve. A slight inequality of balance results from the fact that the disc A must be smaller in diameter than the disc B so as to pass through the valve-seat at B when the engine is being set up. In addition, weights are adjusted to the starting rock-shaft to counterbalance the weight of the moving parts.

In practice the weight of the valves, rods, lifters, etc., is sufficient to cause the valves to seat quickly and firmly enough to give a sharp cut-off. In order to aid the sharpness of the cut-off, however, some builders place springs on the live-steam rods.

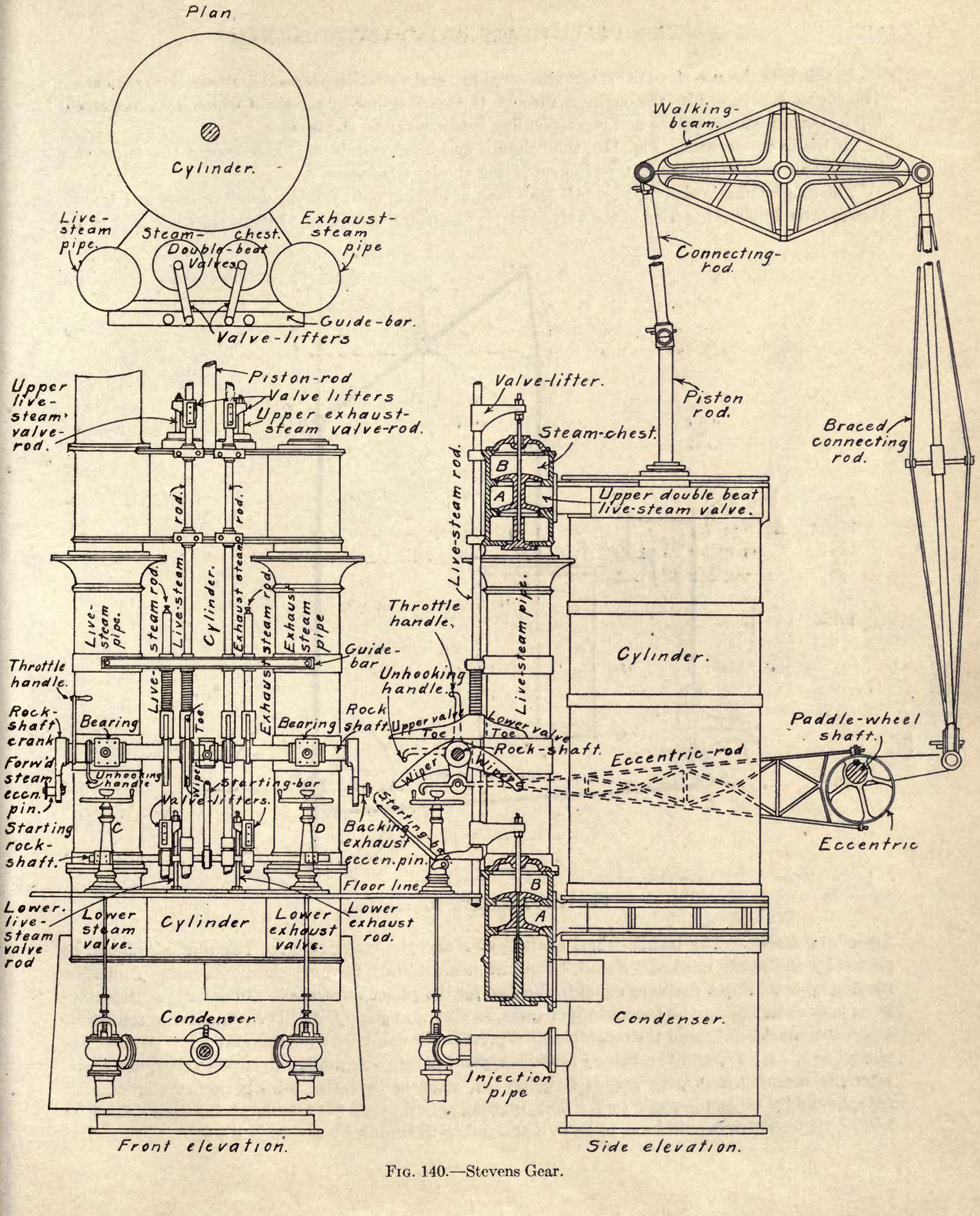


FIG. 140.—Stevens Gear.

In this gear the cut-off position remains constant, and variation of speed is attained by throttling. The engine is reversed by the engineer through the starting-bar by means of which he can open the top or bottom steam-valves and corresponding exhaust-valves at pleasure.

In the front elevation, Fig. 140, the columns and hand-wheels at *C D* represent the connection leading to the valve in the water-pipe supplying the jet condenser.

The diagrammatic sketch, Fig. 141, may help in picking out from the detail of lines in Fig. 140, the essential kinematic action of the valve-gear. The piston is indicated by the dash line at *k* at the

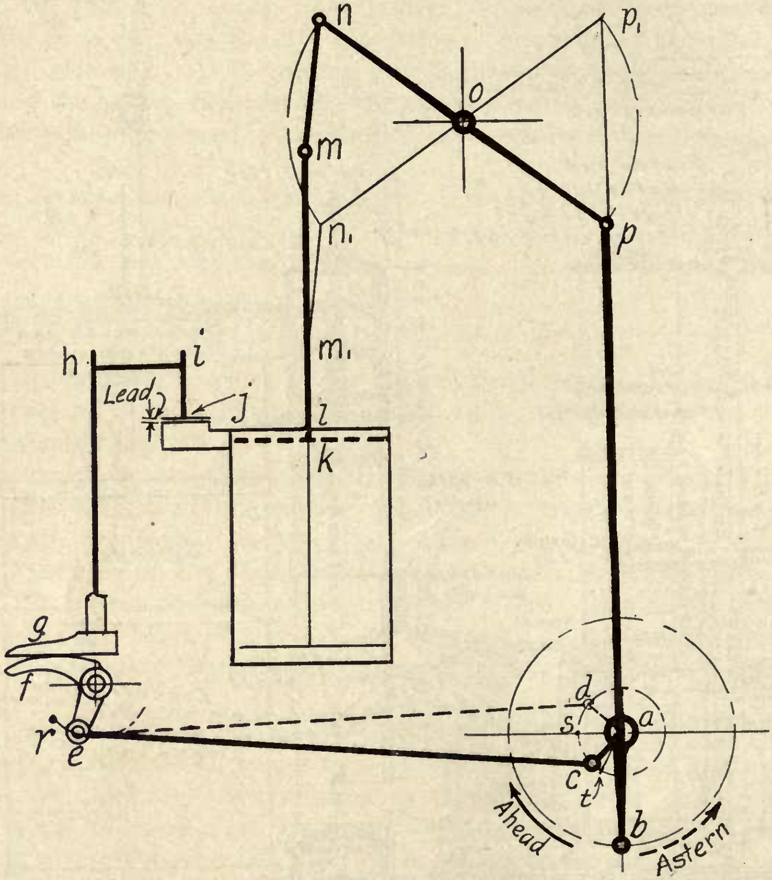


FIG. 141.—Diagram of Stevens Gear.

top of the stroke. The paddle-wheel shaft is at *a* and the crank at *a b*. For running ahead, as shown by the arrow marked "ahead," the piston must start to move down, the valve *j* must be moving up and be up a distance equal to the lead for the phase illustrated. In order that these motions may occur, using the toe and wiper cams, as shown at *g* and *f*, it will be evident that the eccentric center must be at *c* and the eccentric must precede the crank *a b* by the angle *c a b*. When *c* has moved to *s*, *e* is at *r*, and the valve *j* is at its highest point. Assuming that the valve just opens when the eccentric is at *t*, the angle *t a c* being the angle of lead, the total period of admission is represented by twice the angle *t a s*. For running astern, with the engine at the same phase as before, the eccentric would have to be at *d* and follow the crank by the angle *d a b*.

Lentz Gear.

The Lentz engine is of recent date and also makes use of the double-seat poppet-valve, the steam regulation being accomplished, however, by varying the cut-off through a novel type of shaft-governor and straight-slot eccentric instead of by throttling. The first Lentz engine was built in Germany in 1899 and since then four millions of horse-power of this engine have been built and installed in that country. In 1909 the Erie City Iron Works of Erie, Pa., obtained the rights for this engine and have since built and installed over 50,000 horse-power in this country.

Detail views of this engine are shown in Figs. 142–145. The cylinder has four double-seat poppet-valves, one at each corner; the live-steam valve for the crank end is shown in longitudinal section

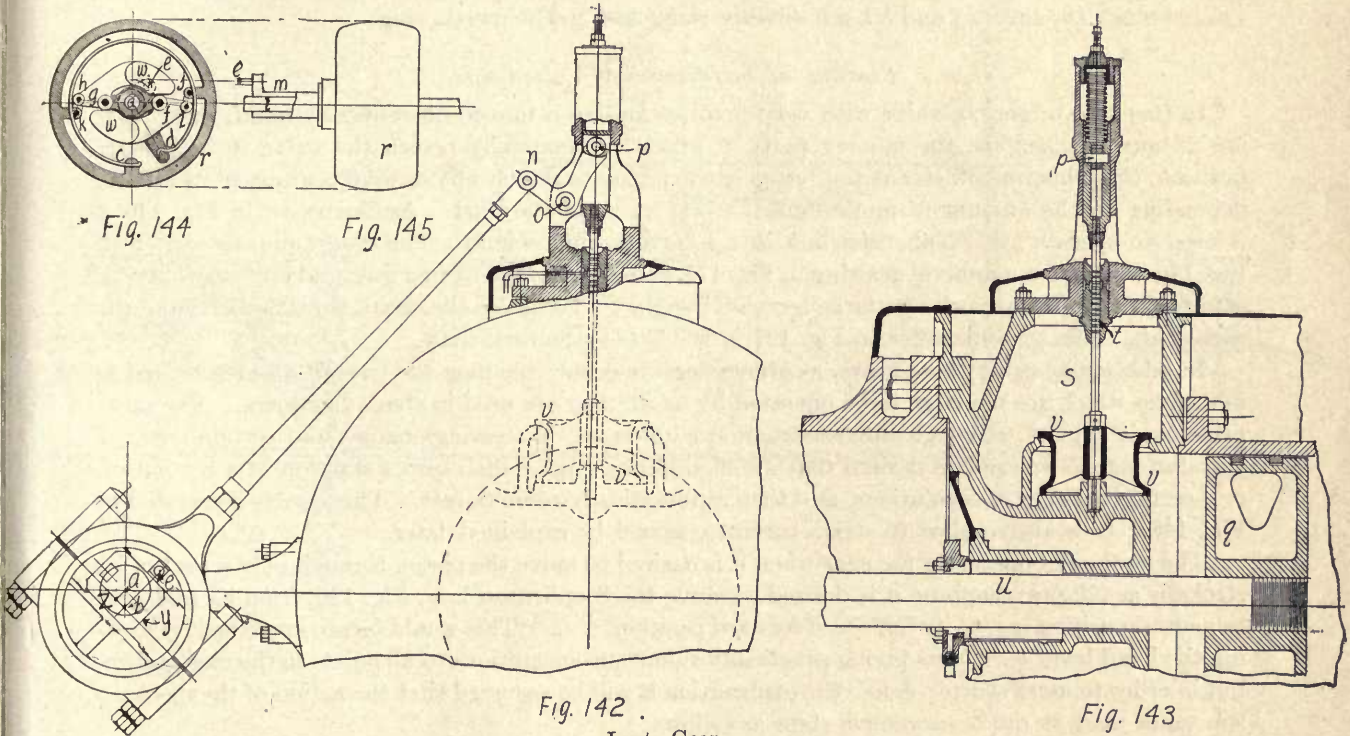


Fig. 144 Fig. 145 Fig. 142. Fig. 143

Lentz Gear.

in Fig. 143. The steam chest is shown at *s* and the valve at *v*. The valve-stem has no packing but is kept tight by the series of turned rings, called water-rings, shown at *t*. Likewise the main stuffing-box at *u* is kept tight by a series of iron rings accurately fitted, no packing being used.

A transverse section of the valve-actuating mechanism is shown in Fig. 142, the valve-stem being actuated by an oscillating cam shown at *p*, acting on a roller attached to the valve-stem. The cam curve is so designed as to disengage from the roller when the valve comes to a seat, but remains in contact until the valve is seated, thus preventing noise and permitting any engine speed. The cam surface is on an arm of a bell-crank *p o n*, the other arm being actuated by the eccentric-rod *a n*. The eccentric-center is at *b* and the eccentric or lay shaft-center at *a*. The eccentric shaft, *a*, runs longitudinally along the outside of the cylinder and gears with the main shaft through a special form of bevel gear. The slide-block, *y*, is permanently keyed to the eccentric-shaft, and the total throw of the eccentric is twice *a b* for the position shown. When the engine speeds up the governor

throws the ***small*** slide-block, e, which works in a small transverse slot in the straight-slot eccentric-sheave, thus moving the sheave itself along the block, y, and changing the total throw of the eccentric to a minimum of $a\ z$.

The action of the governor itself is unique, and is illustrated in Figs. 144 and 145. A one-piece carrier having three arms, $a\ g$, $a\ f$ and $a\ d$, is keyed to the eccentric-shaft. A heavy inertia ring, r, mounted on a hollow shaft, m, which turns freely on the eccentric-shaft, gets its rotary motion through a flat circular spring, $c\ d$. Attached to the hollow shaft, m, is an arm carrying the pin e. As the engine changes load and consequently speed, the inertia ring acts instantly, and the centrifugal weights follow as soon as the frictional resistance of the moving parts is overcome, to move the pin e and the eccentric-sheave along the block y. The governor-weights, w, w, swing on pivots, f, g, and, through the links $i\ j$ and $h\ k$ are directly connected to the inertia ring.

Floating or Self-Centering Valve-Gears.

In this type of gear, a valve with very small steam-lap is moved off center by hand, thus starting an auxiliary engine, the moving parts of which automatically return the valve to its central position, thus shutting off steam and bringing the piston to rest at any desired position in its stroke, depending on the amount of motion given to the valve at the start. As illustrated in Fig. 146 it is used to operate the Stephenson link in a heavy marine engine. The diagrammatic sketch of Fig. 146 is shown in a general drawing in Fig. 147 where the self-centering gear and engine are shown attached to the framework of a large ferryboat engine. To follow the detail construction from the rock-shaft, O, to the link, refer to Fig. 127 in which O is the rock-shaft.

In addition to using these gears, as above described, for changing the cut-off, and for reversing in engines which are too large to be operated by hand, they are used in steam hammers. The same principle is applied, although the construction is different, in steering engines and certain types of elevator engines where it is desired that a self-centering engine shall turn a fraction of a revolution or a certain number of revolutions, and then automatically come to rest. This case is illustrated in Fig. 148. It is also applied to steam turbines, as will be explained later.

The method of operating the gear when it is desired to move the piston through only a part of its stroke is as follows: Suppose it is desired to move the Stephenson link, Fig. 146, from its mid-gear or neutral position, $r\ s$, to its full speed forward position, $r_1\ s_1$. This would be accomplished by moving the hand lever $a\ c\ b$, thus giving practically simultaneous motions to all points in the mechanism, but in order to more sharply define the explanation it will be assumed that the action of the mechanism takes place in quick successive steps as follows:

1. The engineer moves the lever from a to a_1, carrying b to b_1, d to d_1, e to e_1, and the valve from v to v_1, thus opening the port f to steam by the amount $e\ e_1$ less lap. $e\ e_1$ equals steam-port width plus lap. The point n is assumed to remain stationary for the time being. As soon as the engineer brings the hand lever in the position $a_1\ b_1$, he clamps it there, thus securing the end of the short connecting link $b\ d$ at d_1 to act an instant later as a temporarily fixed turning point for the "floating" lever $n\ e$.

2. As soon as the port f is opened the piston moves to the right, driving the crosshead from k to k_1, the rock-shaft lever from $o\ l$ to $o\ l_1$ and thus securing the desired rotation of the rock-shaft and with it the required motion of the link $r\ s$ through the arm $o\ p$ and the bridle-rod $p\ q$. The "return-arm," $o\ p$, may or may not be in line with $o\ l$ according to the construction of the engine and framework.

3. The point m of the rod $m\ n$ is attached to the lever $o\ l$ and is carried by it to m_1, thus causing n to swing to n_1 about the temporary center d_1, and e_1 to swing back to e, again placing the valve on center and shutting off steam just as the piston reaches k_1.

Theoretically there should be no steam-lap on the valve, but this gives too sensitive an action. In practice a very small steam-lap is used—about $\frac{1}{16}''$. The exhaust-lap is added for cushioning and may be about $\frac{1}{8}''$.

If it is desired to move the Stephenson link back from full speed ahead to say half-speed ahead, the engineer would move the hand lever to a point midway between a_1 and a and clamp it there. This would move d_1 halfway to d, e halfway to e_2, and cause the valve to open the port g halfway approximately. The piston would then be driven from h_1 toward h, and would come to rest when it had caused n_1, through the intervening mechanism, to reach a position midway between n_1 and n.

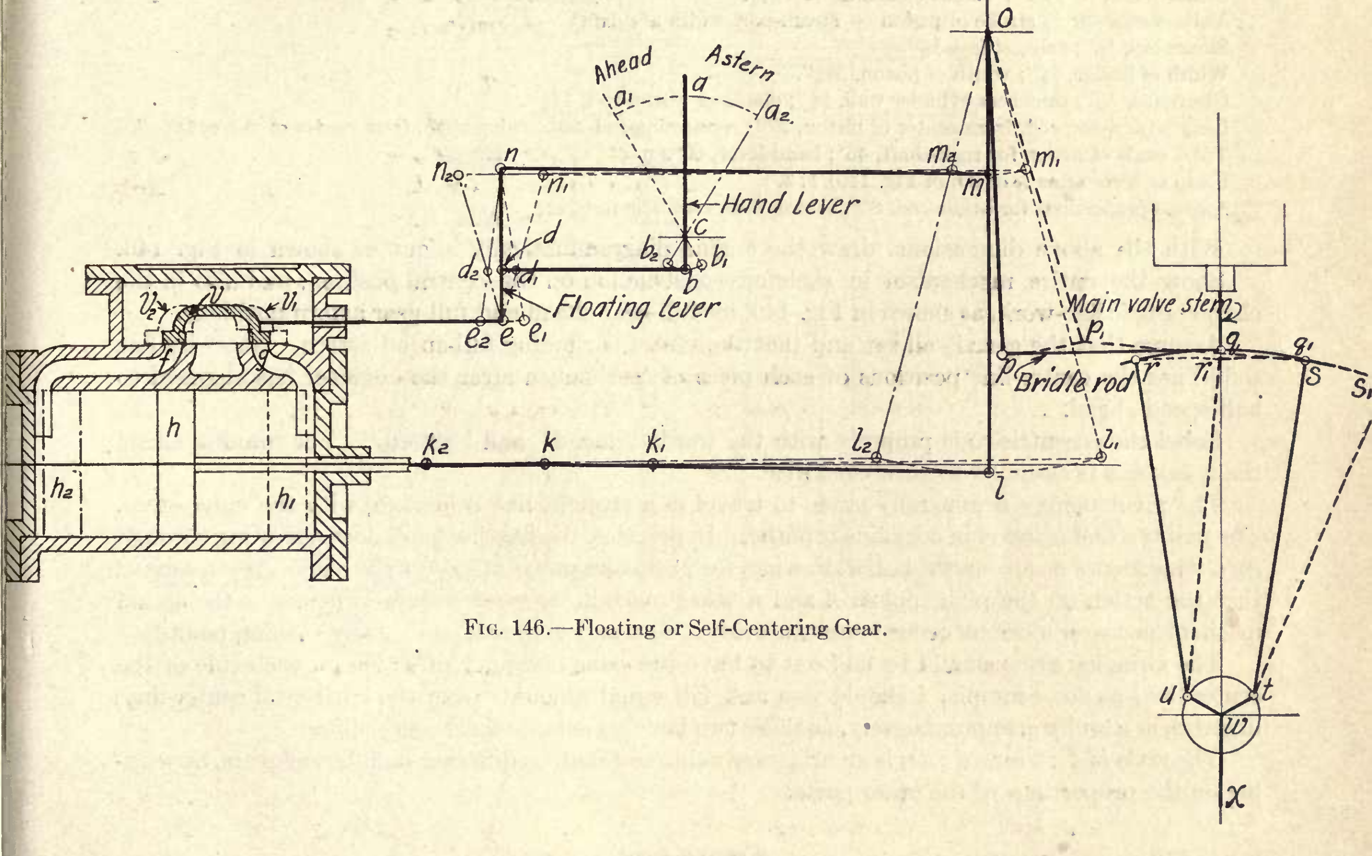


FIG. 146.—Floating or Self-Centering Gear.

The new positions of the pivot points n_1, d_1, and the point e would then all be in one straight line and both ports would be closed with the valve on center.

Should the expansion of the steam, or the weight of the moving parts carry the piston beyond the desired position, the valve would also be carried beyond its central position, thus admitting steam on the other side and quickly balancing and securing the piston at the desired point.

When the crosshead, or some other portion of the mechanism, is not locked at a given running speed, the weight of the piston, crosshead, connecting-rod, etc., especially if these parts are in a vertical position, will cause the entire mechanism, including the Stephenson link, to move gradually until the valve is drawn to one side by the amount equal to its lap, when steam will enter the port and drive the piston and entire gear back to the desired position. In the meantime the slight chang-

ing of position of the Stephenson link has been gradually changing the point of cut-off so that the speed of the engine has gradually increased or decreased to a certain point and then suddenly recovered. This action is technically referred to as "creeping," and its constant recurrence is noticeable in vessels carrying this form of unlocked gear.

Drafting Table Problem, No. 8.

Construct a floating or self-centering valve-gear from the following data:
Cylinder; bore, 9″; stroke, 18″.
Steam-ports, 8″ × 1″; exhaust port, 8″ × 1½″.
Valve-throw for ½ stroke of piston = steam-port width and lap.
Steam-lap, 1/16″; exhaust-lap, ⅛″.
Width of bridge, ¾″; width of piston, 2¾″.
Clearance, ½″; thickness cylinder wall, ⅝″; diameter piston-rod, 1¾″.
Length of piston-rod, from center of piston, 27″; connecting-rod 25″; valve-stem, from center of valve, 16½″.
Total angle of action for rock-shaft, 45°; hand lever, 60°.
Ratio of lever arms (*e d: d n* of Fig. 146) 1:3.
Assume proportions for bridle-rod, Stephenson link, eccentric-rods, etc.

With the above dimensions, draw the engine diagrammatically about as shown in Fig. 146. Show the entire mechanism in skeleton construction on the central position, and also in the characteristic line-work, as shown in Fig. 146, for full-gear ahead and full-gear astern positions.

Assume that the gear is all set and that the vessel is running full-speed astern. Show by fine solid lines the center-line positions of each piece of mechanism after the engineer has changed to half-speed ahead.

Label the eccentric-rods properly with the words "ahead" and "astern." For running ahead the crank *w x* is assumed to turn clockwise.

The pivot point *e* is generally made to travel in a straight line coinciding with the valve-stem. The points *d* and *n* travel in curvilinear paths. In practice, the floating lever does not swing through such wide angles as are shown in the drawing, for in the design, as stated at the outset, it is assumed that the action on the pivot points *d* and *n* takes place in successive steps, whereas in the actual mechanism these motions occur simultaneously. *o*, *c* and *w* are the only fixed turning points.

The swinging arms should be laid out to have the same obliquity of action on each side of the center-line, as for example, *l* should rise and fall equal amounts from the horizontal center-line; likewise, *m n* and *p q*, approximately, as these two have no definite fixed center-lines.

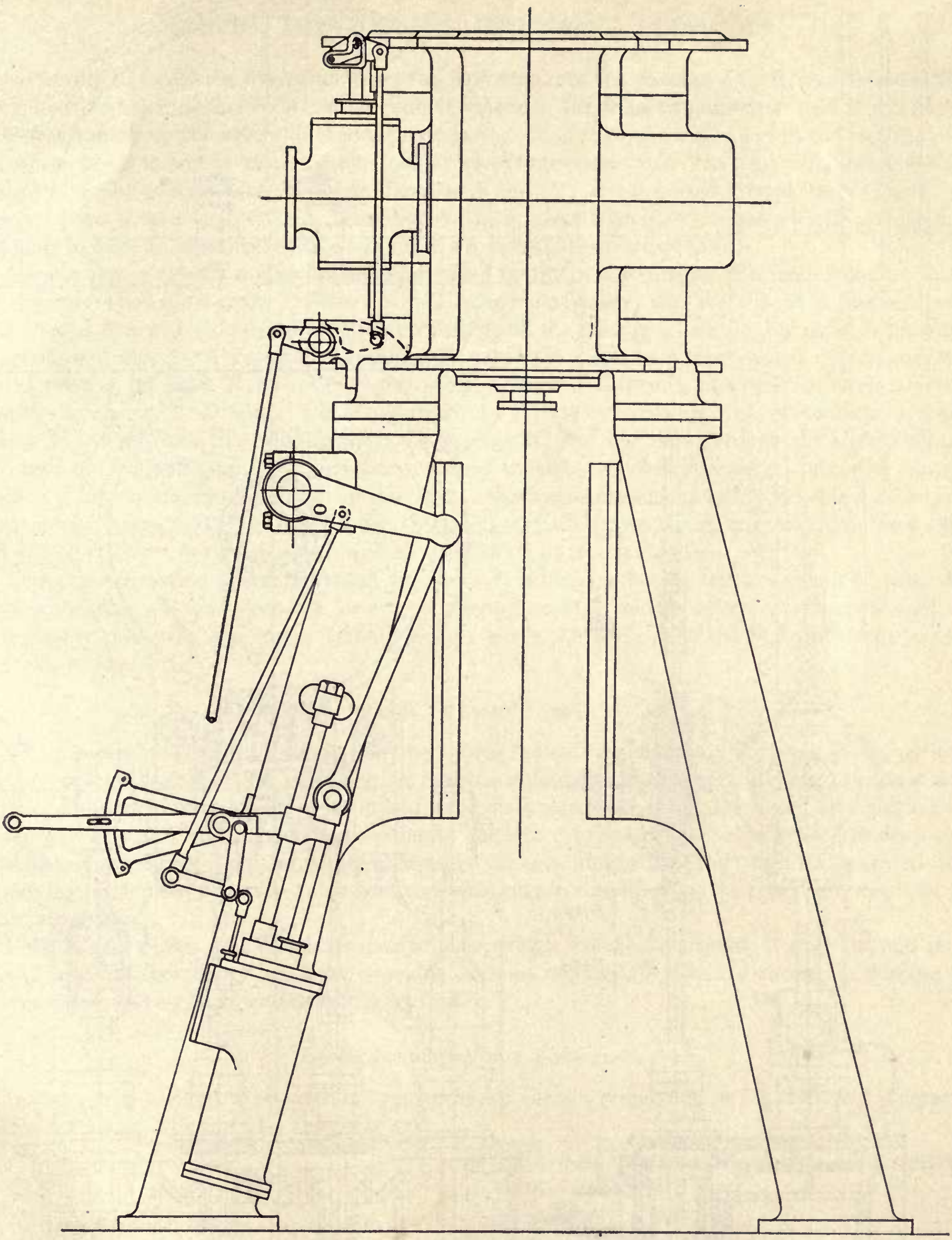
The ratio of 1 : 3 for *e d* ; *d n* is an arbitrary value and may be different in different gears, depending on the proportions of the other parts.

Steering Gear.

The principle of the self-centering valve-gear is here used to obtain a certain rotation of a drum carrying the tiller rope, for a given swing of the pilot's wheel, or, in other words, a certain number of turns of the auxiliary steering-engine for a given motion of the self-centering valve. This is accomplished as follows:

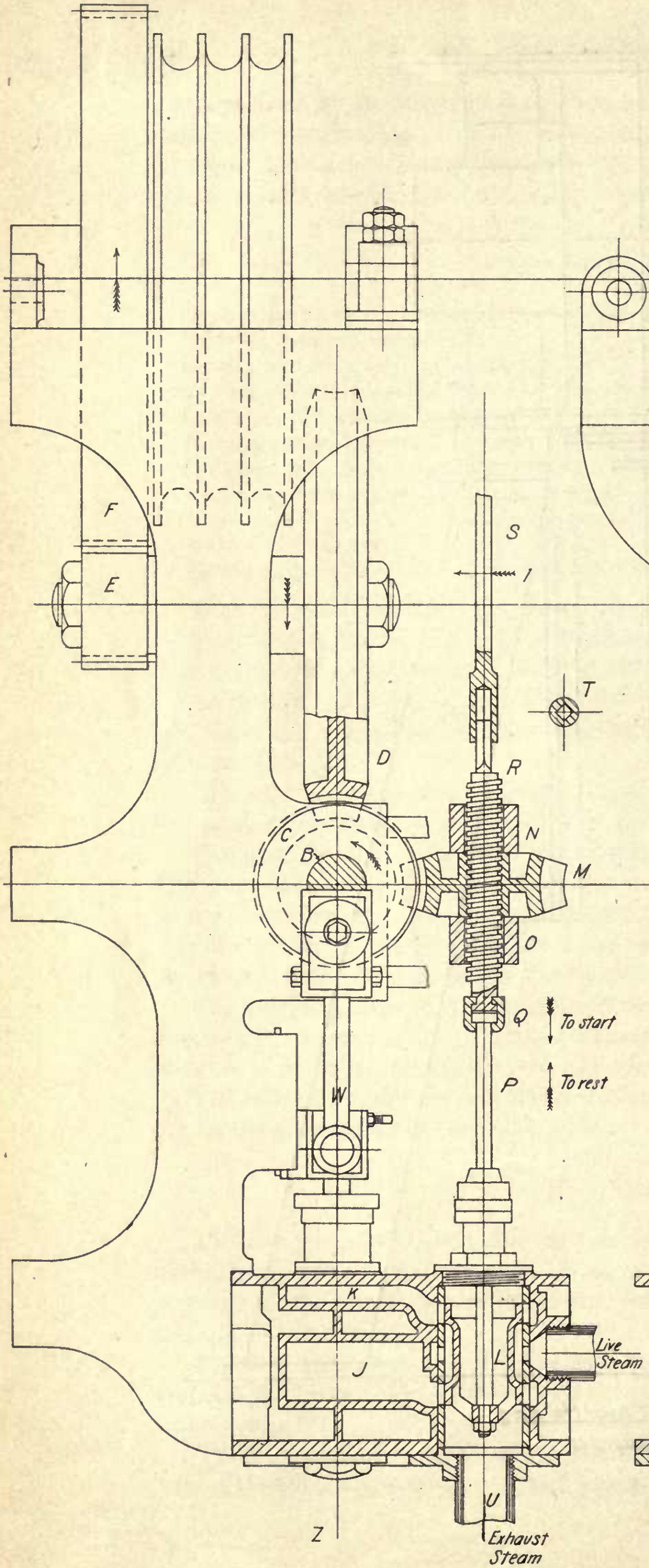
The two engines *A* and *A′*, Fig. 148, drive the shaft *B*, Fig. 149, the power being transmitted through the worm *C* and wheel *D*, and the spur-gears *E* and *F* to the drum carrying the rope to the rudder arm.

The admission of steam to cylinders *A* and *A′* is controlled in the usual way by the hollow piston valves *G* and *H* respectively. The floating or self-centering valve *L* is a third piston valve which



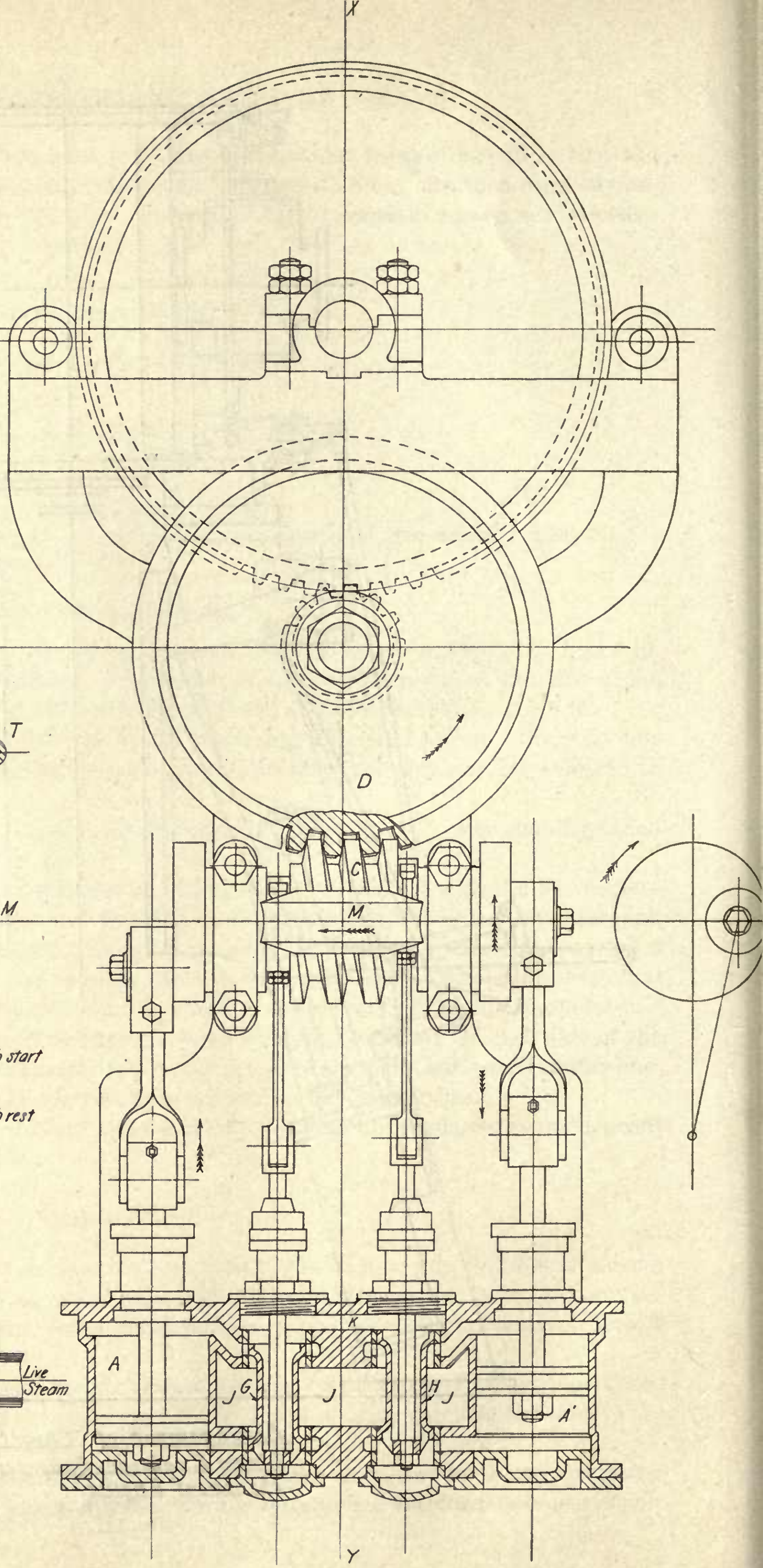
Arrangement of Throttle and Steam Engineer-Gear for Ferryboat Engine.

Fig. 147.



Section through XY.

FIG. 149.



Section through WZ.

FIG. 148.

serves simply to direct the live steam from the feed pipe into the passage J or K, thus establishing inside or outside admission for A and A', and therefore determining the direction that B will have.

Before taking up the action of the mechanism as a whole, attention is called to the fact that M is a worm wheel mounted on a threaded rod R, and is prevented from having longitudinal motion along the rod by contact with the frame shown at N and O. R is connected to the valve-stem P by a swivel joint Q, and to the rod S (connected with the pilot wheel) by a square-section slip joint, as shown in view T, allowing relative motion in a longitudinal direction only.

Assume that a certain motion of the pilot wheel results in the turning of S, and therefore R, in the direction shown by arrow l; worm M will remain stationary, and the thread in the hub will cause R and L to move down, allowing live steam to fill the passage J, and giving inside admission to cylinders A and A', A' being the starting cylinder in the phase here represented. Since the two cranks turning the shaft B are set 90° apart, the engine will always be in a position to start in the desired direction. B, D and F will have directions shown by arrows. The exhaust steam from A and A' will go from the cylinders into the passage K, and will pass through the hollow center of L into the exhaust pipe U, it being remembered that the valve L was placed below its central position by the initial motion given by the pilot. As soon as the engine starts the worm M begins to rotate as shown and by means of the thread in its hub draws the stem P and the "floating" valve L back to its central position and the engine stops automatically.

If the initial motion of S is reversed, the valve L is lifted, allowing the live steam to enter K, thus establishing outside admission for the cylinders A and A', and therefore reversing the motion of the other members, the engine coming to rest when the valve L is automatically dropped to central position.

Steam Turbine Gears.

Steam control in the reciprocating engine requires that admission must take place during periods that are intermittent and that these periods must be definitely timed with the cycle. In the steam turbine, admission is continuous or in puffs in extremely rapid succession, the quantity being varied by the governor and the valve-gear according to the load on the turbine. Except for the details of construction due to the high speed and exacting conditions under which the turbine is operated, the underlying principles or turbine valve-gear construction are simplified by the practically continuous steam admission.

Both in the Curtis and Westinghouse turbines, which will be illustrated, it may be said in a general way that greater or less steam-opening area, as required, is obtained through a floating or self-centering valve-gear operated by the governor.

Curtis Steam Turbine Valve-Gear.

In the Curtis turbine, illustrated in approximately correct proportion in Fig. 145, and diagrammatically in Figs. 146 and 147, the parts are:

A, turbine casing
B, generator
C, governor casing
D, governor beam
E, governor connection-rod
F, floating lever
G, pilot-valve connection-rod
H, pilot-valve (piston-valve with inside admission)
I, piston
J, piston-rod
K, differential connection-rod
L, differential lever-arm
M, differential link
N, rack

O, spur-wheel
P, cam-shaft
Q, cams
R, cam-rollers
S, controlling valve-lever
T, controlling valve-stem
U, poppet-valve and valve-seat
V, cross transmission-shaft
W, first turbine wheel
X, controlling lever-shaft
Y, steam-chest
Z, governor-weights which swing on knife-edges instead of pins.

Oil, under pressure, is used to operate the piston, *I*. The Curtis turbine has multiple admission

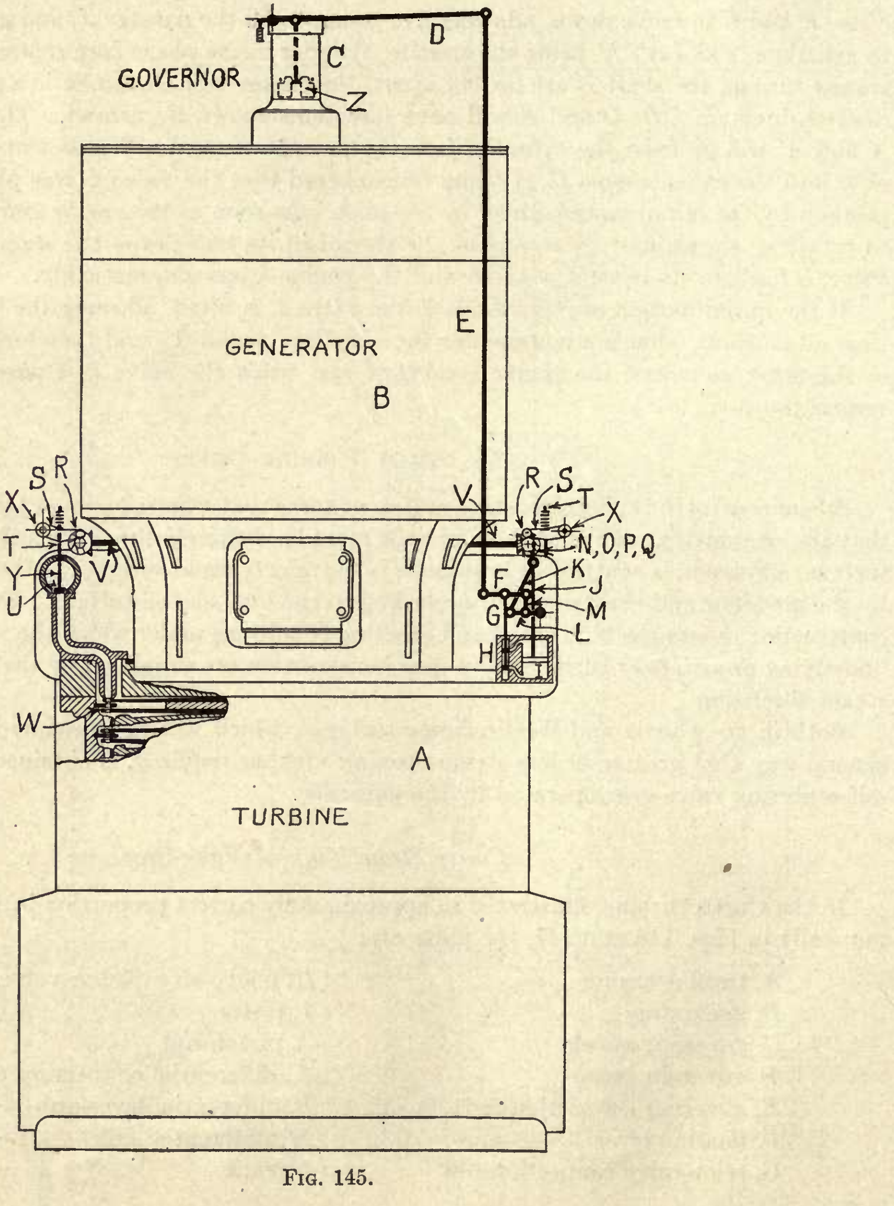


Fig. 145.

valves, each having its own controlling mechanism, Q, R, S, T, and all operated by a single cam-shaft, P. Eight separate admission valves are represented in the top view, Fig. 147. In Fig. 145 two sets of multiple valves are represented, one set on each side of the turbine, both connected by a cross-shaft, and all operated by one governor and one floating valve-gear.

The reason for using multiple valves in this way lies in the fact that all valves that are open

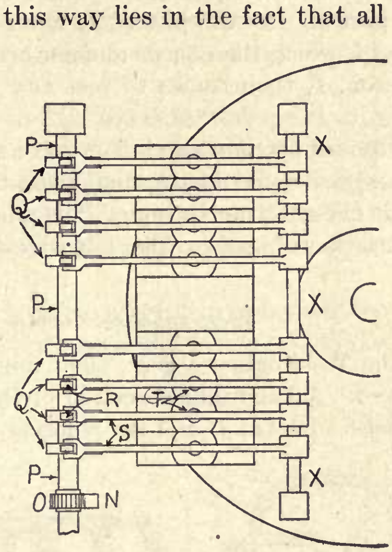


Fig. 147

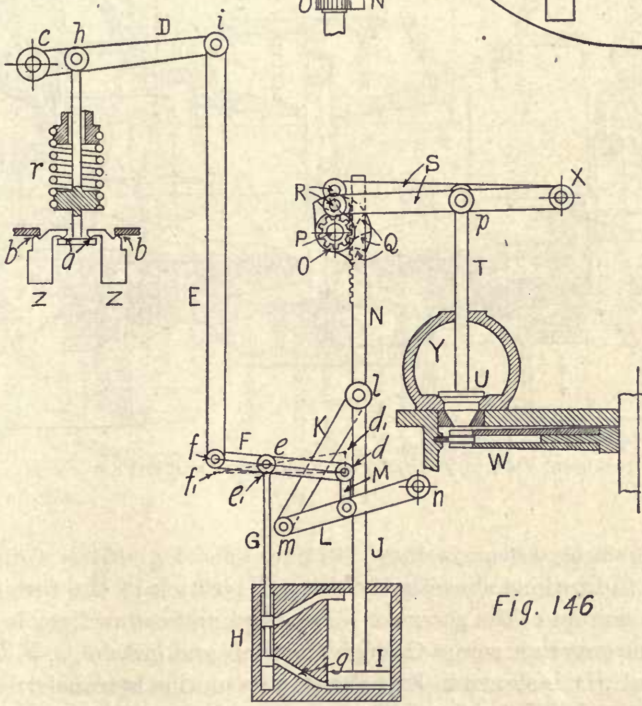


Fig. 146

at all are wide open with one exception and that one is the only one that is throttling the steam. The nozzles thus receive full steam pressure and work to best advantage.

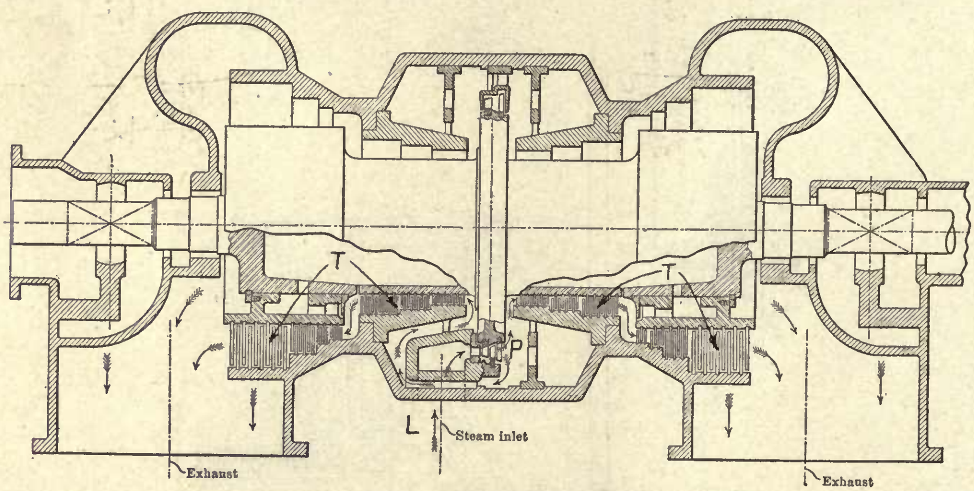
In the operation of the turbine, steam is admitted through a strainer to a combined emergency and stop-valve, not shown in the illustrations, to the steam-chest, Y. When the turbine is at rest

and the governor-weights "dead," all the valves are open excepting the first one, which is just ready to close. As the turbine gains speed the governor-weights, Z Z, fly out, thus causing the governor-beam, D, to turn down about the center c; the point, f, of the floating lever to move down to f_1, turning momentarily about d; the point, e, to move to E_1; and finally the pilot-valve, H, to move down and open the port g. As the piston, I, moves up, the motion is transmitted through the links, K, L and M to d, which then turns momentarily about f_1 thus moving e_1 back to e, closing both ports. The piston, I, then comes to rest and will remain stationary so long as the speed remains constant.

As the piston, I, moves up, one admission valve after another is closed by means of the rack and cams, until the proper speed is attained, when the centrifugal force of the governor-weights balances the tension in the governor spring r. The pilot-valve has a very small lap, thus permitting only a very slight variation of speed. The sensitiveness of the turbine depends on the lap of the pilot-valve.

Westinghouse Turbine Valve-Gear.

For illustrating the Westinghouse gear, their double-flow type of turbine, designed for large powers, will be selected. A longitudinal section of this turbine is shown in Fig. 148. The steam inlet is at L, the impulse wheel at P and the reaction blades at T, T. The housing for the valves,



Sectional View of Turbine, Westinghouse Double-Flow Type.

FIG. 148.

valve-gears, etc., is shown in section in Figs. 149 and 150. Fig. 151 is a diagrammatic sketch of the front elevation of the turbine showing the relative position of the turbine housing, the valve and the governor. A diagram of the governor mechanism and connections is shown in Fig. 152.

The motion from the governor comes through the arms and links m, o, p, w, etc., Fig. 151, to the shaft D. The same shaft, D, is shown in Fig. 150, and its motion is transferred through the arm, E, link, F, and floating lever, J G K, to the pilot-valve, B, and relay piston A. As the turbine changes speed and the governor-weights move in or out, the arm, E, is moved up or down, and with it the point, J, of the floating lever which turns momentarily about the point, K, thus carrying the point, G, and the pilot-valve, B, and admitting oil under pressure from the port, H, to one side or the other of the piston, A, as the regulation requires. If, for example, the piston, A, moves down,

it causes the lever, *t y*, Fig. 149, to turn about *s* and raise the operating piston, *O*, thus giving more steam; at the same time it causes the lever, *K J*, Fig. 150, to turn momentarily about *J*, and so moves the pilot-valve, *B*, back to its central position, closing the oil ports. For any one speed of running the pilot-valve, *B*, covers the ports of admission to the two sides of the piston, *A*, except for a slight oscillation, which is described below, and which allows the oil from *H* to enter both above and below the piston, *A*, at every cycle. The piston, *A*, is constantly oscillating with a very small movement, about 1/8 to 3/16 of an inch.

The phase of the valve mechanism shown in Fig. 150 is for full speed with no load, or in other words, at the instant the load is thrown off and before the pilot-valve has had time to return to its neutral position. The arm, *E*, is full down showing that the governor-weights are at full outward swing. When the turbine is at rest, the governor-weights are at their full inward positions, the arm, *E*, full up, the pilot-valve in its top position with free opening from *H* to the top of piston, *A*, and there is no oil pressure.

Upon starting the turbine the auxiliary oil pump is first set in operation, thus creating a pressure in *H* and driving oil through the open port, *B*, causing the piston, *A*, to descend and thus open

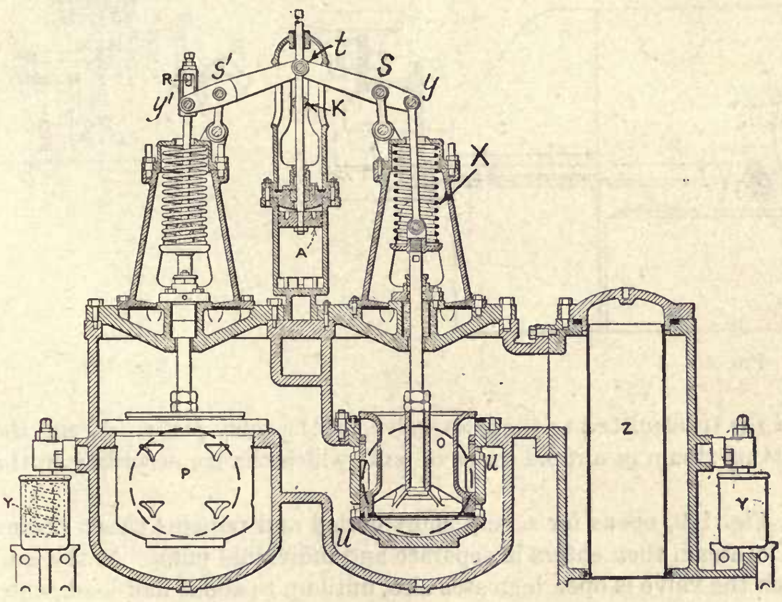


FIG. 149.

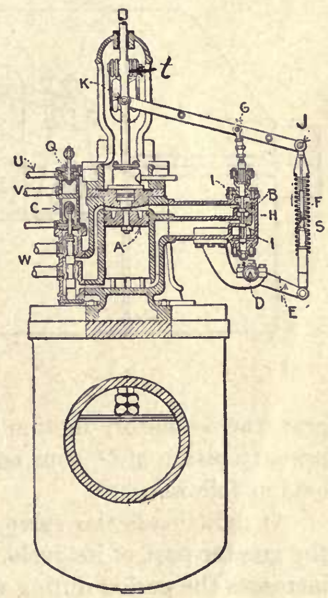


FIG. 150.

wide the primary valve *O*. Steam is then admitted by opening the valve in the steam main, not shown. The governor does not "take hold" until the turbine has reached a speed of about 1400 r. p. m., the rated speed for the turbine here described being 1800 r. p. m. When the turbine is not in use the valve, *O*, is kept to its seat by the spring on the valve-stem. Steam is admitted through the strainer, *Z*, to the operating valve, *O*, and thence to the turbine through the circular opening represented by the dash-line circle also at *O*. The valve, *P*, Fig. 149, is a secondary valve, being designed, by means of the adjustable backlash device at *R*, so that its time of opening may be changed by the operator. It is usually regulated so as to open when the primary valve has reached its maximum opening.

Inasmuch as a governor must first absorb energy sufficient to overcome the friction of rest of

the various movable parts of the valve-gear which remain stationary with respect to each other for any constant speed, it cannot instantly transmit the regulating motion for which it is designed. In order to reduce this delayed action, the Westinghouse gear includes a vibratory motion illustrated in Fig. 152, where it may be seen that if *i* were a fixed pivot, the lever, *i j*, would always rotate about the exact point, *i*, for each change of speed, and the connecting link, *i h*, would not be required. At constant speed the governor-weights, *r r*, would then hold the lever, *i j*, stationary through the connecting pin at *f*; also, all the mechanism to and including the valves would remain stationary.

If, now, the point, *i*, Fig. 152, is given a slight vibratory motion by an eccentric, *c*, the lever, *i j*, will oscillate about the point, *f*, when the turbine is running at constant speed, and the valve-gear mechanism will be in constant motion and therefore more sensitive. *a* is a worm keyed to the main shaft of the turbine rotor.

In the older types of turbines of this manufacture, using steam instead of oil for operating the

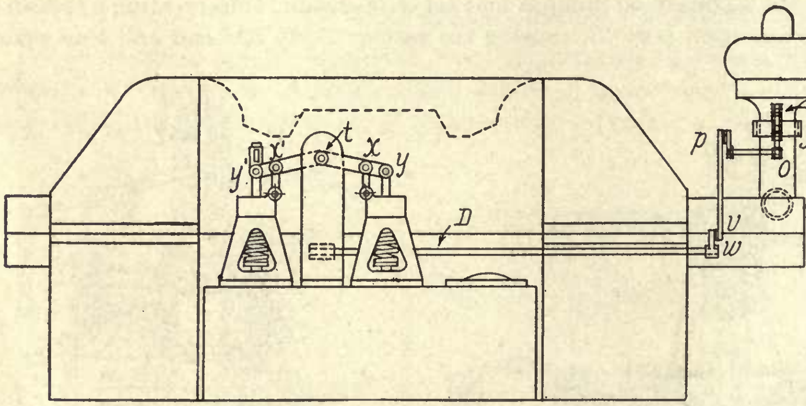


Fig. 151.

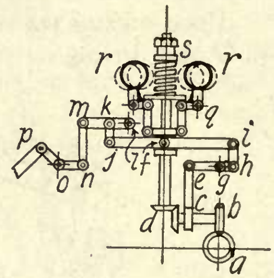


Fig. 152.

gear, the oscillatory motion of *i j* is transmitted to the pilot-valve, *B*, the relay piston, *A*, and the primary piston at *O*, thus admitting steam in a rapid series of puffs which change according to the load as follows:

At light loads the valve, *O*, Fig. 149, opens for a very short period and remains closed during the greater part of its cycle. The steam then enters in separate and individual puffs. As the load increases the period during which the valve is open increases also, until up to about half-load, when the valve ceases to close entirely and the steam begins to have continuous admission to the turbine although the quantity is not constant, as the valve is vibrating near its closed position. When full load is attained, the plane of vibration of the valve is farther away from its seat and the steam enters in practically a steady blast.

In all the larger sizes of Westinghouse turbines the oil relay system is now being used, and the mechanism so designed that the valve, *O*, is held practically stationary in its running position so that there is very little if any fluctuation in the steam pressure entering the turbine.

The valve, *O*, is a combined poppet and piston-valve, acting as the former type when it is closed or nearly so, and as the latter type when it is well open. An auxiliary safety steam valve is shown at *Q W*. It is under control of a speed-limit device and has nothing whatever to do with the regular running of the turbine. The spring at *X*, Fig. 149, is used to close the primary valve, *O*, in case of failure of oil supply, and to keep it closed when the turbine is not in service; the spring at *S*, to

relieve the mechanism of strain. These are details of construction which are not essential to a general understanding of the turbine gear when running under regular conditions. It may be explained, however, that with the turbine at rest and oil pressure removed, the spring, X, will keep the valve, O, closed and the piston, A, at the top of its stroke. Also, with the turbine at rest, the governor-weights will be full in, and the lever, E, at its highest position, which would cause a strain on the link, F. This strain is taken up by the spring, S. In case of failure of oil supply while the turbine is running, the spring, X, will close the main valve.

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